Cal Poly FormulaSAE Engine Development

Sponsor: Cal Poly SLO Formula SAE

SPEED Systems

Matthew Ales - males@calpoly.edu

Rafael Mendoza - rjmendoz@calpoly.edu

Mitch Thomas - mthomas@calpoly.edu

Leon Vinokurov - lvinokur@calpoly.edu

Mechanical Engineering

California Polytechnic State University, San Luis Obispo

June 2010

Table of Contents

Table of Figures

Table of Tables

Executive Summary

This project, engine development, was sponsored by Cal Poly FormulaSAE. The FormulaSAE team proposed this project in order to have a more reliable and more powerful engine for their car, to improve their overall performance at their competition. The baseline output of the engine is 37 hp and 24 lb-ft of torque. The goal of this project was to increase the output to 45 hp, to meet requirements determined by a trade study. We planned to increase the engine's output using several strategies.

We evaluated all potential design decisions using a Ricardo WAVE model. Ricardo WAVE is a software tool that can simulate the operation of engines and their components. Using accurate measurements of important engine parameters, WAVE simulated the results of engine dyno testing, allowing us to narrow down our design choices quickly and inexpensively. The model predicted a baseline of about 40 hp and 30 lb-ft of torque. When comparing the WAVE model and the baseline dynamometer test of the engine for model verification, the power curves were similar but with the simulation curve being shifted toward higher engine speeds. The model predicted higher power because the model shows power at the flywheel while the dynamometer measures power at the brake, after the gearbox and sprockets. The model was also used to predict the effects of increasing the engine's compression ratio as well as the effect of different camshafts and valve timing.

By carefully determining gear ratios and rear sprocket size we allowed the car to have as much acceleration as possible in each driving event. The amount of force that the car transmits to the ground depends directly on the gear ratios in the transmission and the ratio of the engine sprocket to the axle sprocket. Thus, gearing is an important aspect of overall car performance and is set up to match the powerband of the engine.

In order to increase power the camshaft profile and timing were both altered in order to provide a powerband more suitable for the FormulaSAE competition. The current powerband is designed for the high revving endurance motorcycle where top speed is critical and the intake is unrestricted, while FormulaSAE places a premium on power at lower speeds to help the car accelerate out of corners. Another major reason for bringing the powerband down lower is that

at lower engine speeds, the effects of the restrictor are minimized when compared to higher engine speeds.

The engine intake directly affects the ability of the engine to intake air. More air in the cylinder means more fuel burned and more power, so devising a way to maximize the amount of air in the cylinder is an excellent way to increase engine power. As the engine is essentially sucking air in from the atmosphere, nearly anything that reduced head loss in the intake tract was beneficial. Wave dynamics were equally, if not more, important in intake design. By careful analysis of pressure wave dynamics, the intake tract was optimized for increased power at a chosen engine speed.

As with the intake system, the exhaust system focused on reducing pumping losses and harnessing wave dynamics to increase engine performance at a specific engine speed.

All of these efforts together give the FormulaSAE team the performance it needs to perform well at the FormulaSAE competition in Detroit. As a result of our efforts, the engine makes 42 horsepower at peak, which is a 13.5% increase. The engine's peak torque is now 27.5 lb-ft, which is a 14.5% increase. The engine also has a 10% increase in power, as well as a 12.5% increase in torque across the powerband, as we shifted the powerband to a lower engine speed.

1. Introduction

The FormulaSAE engine development project is proposed by the Cal Poly FormulaSAE team to improve the overall performance of the current engine. The current engine being used is the single cylinder WR450 from Yamaha. As with all racing engines better performance is beneficial because a more powerful engine gives the team a better chance to succeed. We will help FormulaSAE achieve their goal of gaining a top 10 finish at the FormulaSAE competition by improving the performance characteristics of their engine. In order to meet this overall goal, we will set and meet several smaller goals. We must increase the power-to-weight ratio of the engine, and increase the area under the torque and power curves to improve general drivability while maintaining or improving its reliability and keeping cost low.

Figure 1: A Formula Car on a Chassis Dynamometer

In order to achieve success, each subsystem must be reworked and tested in concert with the other subsystems. The best combination will allow for getting as much power and torque out of the engine as possible.

2. Background

The FormulaSAE competition is a collegiate competition where schools from around the world compete in both static and dynamic events judged by industry professionals. The rules for the FormulaSAE competition are very broad but require the students to design and build an open wheel style race car with a minimum 60 inch wheelbase and a four stroke gasoline engine displacing a maximum of 610cc. The static events include a design event where students discuss and justify their designs, a cost event where students discuss the cost of manufacturing the car and a marketing presentation where the students are to pitch their design to a group of

Figure 2: All the competitors at a FormulaSAE competition.

investors looking to produce the vehicle. The dynamic events include a skid pad, acceleration, autocross and 26km endurance course. Each event has a maximum point value and the team with the most points at the end of the competition wins.

The rules regarding engine are simple but pose large challenges to the competitors. As mentioned, engine choices are limited to 610cc maximum displacement generated by a four stroke engine running on 93 or 100 octane gasoline, or E85 ethanol. In addition a 20mm restrictor (19mm for E85) must be placed before the inlet to the engine but after the throttle body. The restrictor is the largest challenge for the engine as it forces the teams to create their own intake and fuel injection maps to minimize its effect.

Figure 3: The WR450 Engine on the Dyno.

The 2009 Cal Poly FormulaSAE car, named CP09, was a 336lb car using the single cylinder WR450 producing approximately 37 hp. In past years the FormulaSAE team has focused on converting the carbureted WR450 engine to fuel injection per competition rules. This includes a custom fuel and ignition map to extract as much of the engine's potential as possible. Because the engine starts as a carbureted engine, there are no baseline fuel or ignition maps to use as a starting point. For this reason, producing a baseline map that will start and run the engine is very difficult and takes a lot of trouble shooting and repair time. For the 2009 competition the team was able to create a fuel and ignition map which produces a moderate amount of power and starts the engine most times. With the focus of the team on the map portion of the engine, very little time has been spent on developing other parts of the engine subsystem. As a result, intakes and exhausts have been built and placed on the dynamometer but not engineered to the standard needed to win the competition. In addition, minimal time has been spent making the engine reliable. The engine has oftentimes been the most unreliable part of the car over the past 4 years. With the team's entire focus placed on getting the engine to run consistently very little long term testing has been completed.

The most popular engine type in the FormulaSAE series is a 600cc 4 cylinder. These engines can easily be tuned to 75 hp or more at the wheels with a typical vehicle weight of about 500 lbs. The 2004 Cal Poly FormulaSAE car incorporated a Yamaha R6 engine producing 75 hp at the wheels with a low total weight of 440lbs. To be competitive in acceleration and all events the current car must have a high power density (power per unit engine weight) to maintain a competitive power to weight ratio.

In each of the last four years, the top three teams at the FormulaSAE competition in Detroit, Michigan, where the team plans to attend competition this year, have almost all run a four cylinder, 600cc engine, that has been developed for many many years. In order to produce a top finisher we must overcome the WR450's power and reliability disadvantage, making the success of our project essential to the team's success.

The FormulaSAE team performed a trade study using a proprietary lap simulation to compare power output of the engine to the number of points received in each event. They determined from this study that to achieve a top 10 in the most difficult competition, FSAE in Detroit Michigan (2007) their engine would need to output 49 hp, but to achieve a top 10 result at the 2008 FormulaSAE West competition their power requirement would be only 42 hp. As a result we picked a number in the middle, 45 hp, for our minimum power output goal with the additional goal of improving not only peak power, but making gains across the powerband.

3. Objective

Our engineering target was 45 horsepower peak, without excessive weight gain. As weight gain might have been unavoidable, we considered it acceptable so long as the power to weight ratio of the car as a whole increased. Our other objectives were a power increase throughout the entire powerband and no decrease in engine reliability and maintainability.

To improve power we developed a power curve that best fit FormulaSAE's application. The car is a very light autocross vehicle designed for a novice driver. As a result a "peaky" powerband is not ideal, so we will try to broaden it. Further, being an autocross car, acceleration out of corners is more useful than high maximum vehicle speed given an infinite amount of space, so a wider usable powerband would again be more important than higher peak power. In short, we will develop a wider, more suitable powerband with a peak of 45 horsepower. More area underneath the power curve will greatly improve overall drivability.

Figure 4: The Cal Poly FormulaSAE Car

The most crucial aspect of any race car is reliability, since a car that cannot start cannot win. In the 30 minute endurance race competition, the car must be able to start and run consistently or it will not finish the race. Thus another of our goals is to improve the overall reliability of the engine through large amounts of testing and tuning. Over the past several years, the FormulaSAE team has had numerous reliability issues with their engine. Much of this can be attributed to incomplete tuning. This season the engine proved much more reliable as a result of more involved tuning but it is still a subsystem of the car that is constantly worrisome. Unfortunately, reliability and the quest for more power are often mutually exclusive.

With any race car, problems occur unexpectedly. These problems require quick solutions because time is at a premium in competition. Any time spent repairing the car directly affects the team's chances to compete in dynamic events. Accordingly, our third goal is improved maintainability. Whatever time the team can save in the pits through a more easily maintainable engine is more time on the track accruing points toward the overall standings.

A summarized table of requirement appears below.

Table 1: List of Engineering Requirements for the FormulaSAE Engine Development Project

Accessibility	10 Min Engine swap. Minimize restricted access to other components.
Drivability	Must be able to increase lap times by a considerable amount
Power	45 hp Peak through the use of intake, exhaust and valve tuning.
Power Curve	Lower Powerband in RPM range. More suitable to FormulaSAE
Reliability	Ability to run at least 20hrs without engine rebuild or part replacement.
Weight	Maintain Current Subsystem weight of20Lbs

4. Method of Approach

To meet the desired goals we had a brainstorming session to get all possible ideas down on paper. Once enough ideas were generated, we sat down with the FormulaSAE Team lead, Josh Roepke, and current FormulaSAE engine lead, Cody Scott, to create a decision matrix (Table 16). The decision matrix contained all the possible options for modifying the engine. We then added all the requirements that our team needed to meet. The requirements were reliability, power increase, low weight, low cost, and feasibility to complete. We then weighted these 5 criteria 1-5 with 5 being the most important to the design. Once all the requirements were weighted we then ranked each component on how well it met each requirement, again on a 5-point scale,. We then multiplied by the weights and summed up the total (see Figure 67). The four with the highest point value were to: design new intake and exhaust tracts, modify the engine for higher compression and develop a new valve lift profile.

We began by performing extensive research to familiarize ourselves with the design of the current engine and its components. After establishing a baseline of how well the current components work, we began designing our own systems that will optimize engine performance toward our goals.

The first step of evaluating these designs was creating a Ricardo WAVE model. The effectiveness of different intake, exhaust and cam designs will be modeled. To build an accurate model we will need to spend time measuring various aspects of the engine, such as tract lengths and diameters. In addition to finding out what works best, this process will also show how the engine reacts to the various components.

We will then determine several plausible designs for each subcomponent then test each to determine the best one. We plan to use a dynamometer (dyno) for testing to determine which combinations of designs is the best. Our goal is to achieve 45 hp and 40lb-ft of torque as measured on the dyno. Also, dyno testing will be needed to re-tune the ignition and fuel maps with the new components before they are installed on the car for competition. Once dyno tuning is complete, the engine will be installed on the car and tested. On car testing is necessary to see how the components respond to transient conditions that will be seen in real driving but are difficult to simulate on the dyno. (Appendix A: Decision Matrix, QFD and Management Plan)

To achieve the desired horsepower and torque gain we designed an intake, exhaust, camshaft and combustion chamber specifically made for this car to minimize the effect of the 20mm restrictor. All of the parts interact with one another and are all interdependent in terms of performance gains. The gains from the individual parts cannot be superimposed onto one another and will require individual as well as collective testing to determine total system gains.

We used resonance tuning on the intake and exhaust to improve performance by using pressure waves in these systems to force more air into the cylinder and then suck it out more efficiently. Intake and exhaust tuning are similar in harnessing pressure waves originating from the valves, but may not work to their full potentials if both are designed for the same engine speed. Thus our exhaust and intake tracts will be set up to complement one another and spread their gains over a larger operating range. We chose to tune the intake for around eight thousand rpm and the exhaust for about five thousand rpm in order to widen the torque curve in the operating range of the engine.

Cam profile and valve timing are crucial for getting the best results from the engine. We bought adjustable sprockets, which were essential for testing valve timing. From dyno data it can be determined that the stock camshaft from Yamaha is designed well for a high-revving dirt bike as the engine makes over 40 hp between 7-12k rpm. This design does not translate well to the FormulaSAE competition, since the restrictor required by FormulaSAE rules limits airflow, and thus lowers the maximum useful speed of the engine. We also designed the power and torque curves with a custom camshaft profile. We purchased a set of Hot Cams aftermarket camshafts which matched our design requirements for lift and duration. This was much more cost and time effective since making camshafts is a very specialized process.

The final way we plan to improve horsepower and torque is to increase the compression ratio of the engine. This can be achieved because of the relatively lower volumetric efficiency caused by the restrictor in addition to running on 100 octane gasoline. The chosen piston raises compression from 12.3:1 to 13.5:1 and it was chosen over a 14:1 piston for reliability purposes. We did not want this engine to require frequent rebuilds, and the lower of the two “high” compression ratios gave a larger safety margin when tuning and running the engine.

The metric for determining the success of the final design include peak torque and power, 75 meter acceleration time and points scored in each event, as well as others. Once we completed our QFD analysis of all these components we found that our 75m acceleration time would be the most effective method for measuring success of our design. This made sense because acceleration is directly affected by the power to weight ratio of the vehicle. If power increased while maintaining the same or less weight, then the car would be able to get to speed faster thus gaining the team points.

5. Management Plan

FormulaSAE engine development was a large project that involves the development of several different subcomponents. The plan of attack for our project was to split up each subcomponent and work on them individually and as a group to develop the optimal designs. By working on them individually, each subcomponent had one person focused solely on that component. As a result, that one person focused their attention on making their component as well-engineered as it could be. At the same time, we as a group constantly reviewed all the components to ensure that they worked together to best meet FormulaSAE's goals.

The bulk of this project involved dyno testing. There are many inherent problems with dyno testing and tuning. As a result both SPEED Systems and the FormulaSAE team wanted to have as much testing time as possible. Because the dyno needed to be run by a FormulaSAE team member, dyno time was an extremely limited resource. To combat this we planned to have all our components ready to go on the dyno by the first week of spring quarter. There were many fitment issues and because the engine, like all single-cylinder engines, is essentially a shake table, the weakest link in the system broke on a regular basis. In an ideal world testing could have been completed in approximately 3 weeks working multiple days a week, but because of the tendency for engine components to break we planned for triple that amount of time.

Manufacturing time for this project was fairly short. Because the two most manufacturing intensive components must be ordered from outside vendors, the manufacturing time was a mere 6 weeks. To be completely ready to test in the third quarter it was critical that all our components were ready to go by spring break. This short manufacturing time allowed us an extra 4 weeks during winter quarter to perfect our designs.

The items that had large potential to delay us were the third quarter testing plans and the ordered parts. The ordered parts could have had long lead times. By finishing the component design early we ordered our parts sooner so as not to delay testing if they were not available. From April to June there was a lot of dyno testing completed, consisting of switching components on the engine on a weekly basis. In order to complete all our testing on time we had to work hard and stay on schedule.

6. Design

The Cal Poly FormulaSAE engine development project is broken up into five main sections. Gearing, Camshaft, Intake, Exhaust and Combustion Chamber design. Each one of these areas can be addressed individually though they interact heavily with each other.

6.1 Ricardo WAVE Model

The WAVE model is a great preliminary tool to evaluate the plausibility of all the proposed designs. In order to get a baseline of the engine's performance, the initial model represents the current engine. The most important parameters represented in the model are the intake and exhaust, including the runners within the cylinder head. Many other variables are important to the model such as bore, stroke, air-fuel ratio, spark timing, and other engine parameters.

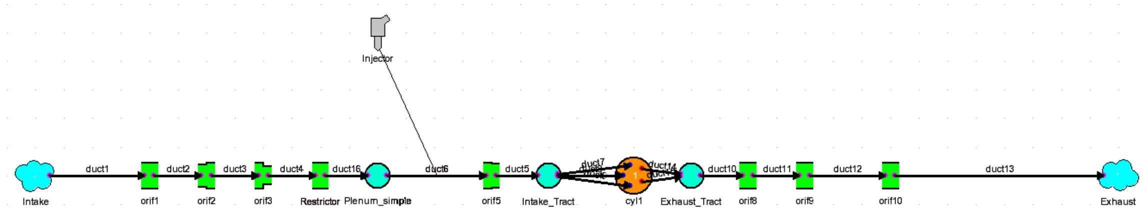


Figure 5: Ricardo WAVE model representation of the engine.

However, the intake and exhaust modeling is the most important step in creating a representative model because the geometric measurements taken from the physical engine are entered into the program and the performance of the virtual engine highly depends on getting accurate measurements.

Since accurate measurements are critical to proper model performance, a disassembled WR450 cylinder head and the current intake and exhaust were measured with dial calipers. The necessary measurements made included the lengths, widths and bend angles of every transition in the intake and exhaust systems. The camshaft profile was measured using a CNC machine and a dial indicator. The lift was measured at every degree of camshaft rotation. Accurate measurements will result in a model that correctly predicts performance of the current engine and performance when modifications are made. Valve timing is likely the most critical part in order to have a representative model in WAVE and it's likely the toughest one to get measurements of. The measurements of the stock cams give a very long duration of 320 degrees at zero lift, and maximum lift of around 0.345 inches. Using these measurements in the model results in low readings, thus the duration was decreased when implemented in order to account for measurement error and the assumed flow coefficient modeling in WAVE. The decrease in duration only affected low lift duration, as duration at 0.050 inches of lift was kept as measured. The simulation was affected by the long duration measured at very low lifts (below 0.050 inches) because the flow coefficients used by WAVE assume that the valves flow better than in reality. The assumed flow coefficients used in WAVE attempt to predict flow past the valves, but they cannot exactly model this. Therefore, the duration at low lifts was assumed to be smaller than measured and was decreased to less than 300 degrees, with duration at 0.050 inches staying at about 245 crankshaft degrees in order to account for inaccuracies in measurements and the assumed flow coefficients in WAVE.

Fuel delivery is accurately modeled in the program as a "pulse-width injector." The actual table used in the MoTec control unit on the car was used to input the fueling curve into the software. The wide-open throttle pulse width was used in the model since the throttle is not modeled, thus it represents a full-throttle dynamometer test. A spark timing model is also not included but is unnecessary because Ricardo WAVE uses a "50% burn point" which is the location in crank degrees where the mixture is half-burned. Since the burn point stays fairly constant, due to changing the spark timing, over the engine's speed range a variable burn point is not needed to achieve an accurate model.

After all the necessary parameters are incorporated into the virtual engine it will enable tuning of the system within the simulation, reducing the need for testing of the physical engine. A change in the model will show how performance changes with a modification and those trends should accurately predict how the actual engine will react. As long as the measurements are correct, the simulation will predict the performance of the physical engine very accurately.

Once the model can accurately predict how geometric changes affect engine performance it can be used as a powerful tool in deciding on the most worthwhile modifications. It is also a great tool for optimizing a certain design by showing how performance changes by varying a certain property of an intake or exhaust tract, such as length or diameter. The WAVE model enables a theoretical design to be simulated before anything has to be fabricated, fitted and tested on a dynamometer.

One fact that must be noted is that WAVE predicts power at the flywheel, and the dyno measures power at the brake. There are power losses between the flywheel and the brake that must be accounted for. WAVE features a calculation of drive train losses but not necessarily for this project's specific application. Therefore, WAVE results should be assumed to be slightly high when compared to dyno readings.

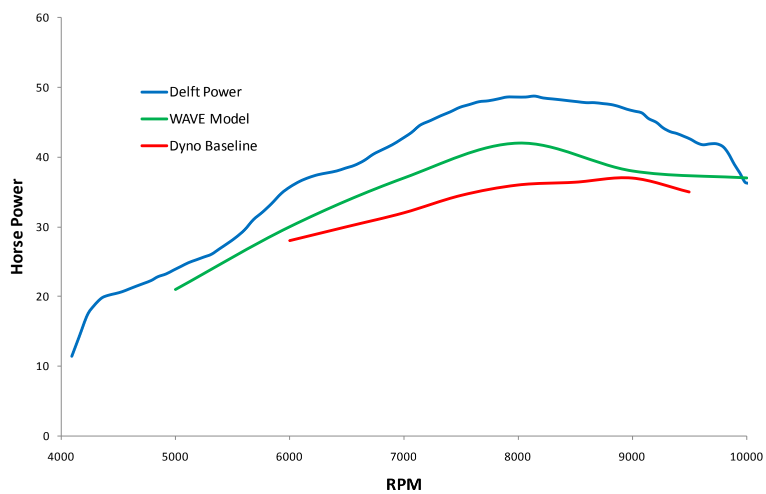


Figure 6: Power comparison of TU Delft, WAVE Model, and Cal Poly's baseline.

In order to validate the accuracy of the model, the stock engine was tested on the dyno. A valid model will properly predict the results of changes made to engine parameters. The real engine's valve timing is a parameter that can be altered with an adjustable camshaft sprocket. A change in valve timing can also be quickly achieved in the software. Figure 6 shows a comparison between the current engine, WAVE model and the results of a dyno test performed by Delft Technical University of the Netherlands of the same engine model. The WAVE model shows slightly higher power production but this can be accounted for by the losses induced during the power transmission between the flywheel and the water brake on the real dyno. Otherwise, the trends shown in Ricardo are similar to what the dyno produces as well as TU Delft, albeit Delft has clearly put a good amount of development into their engine. With some more development of the WAVE model, including up to date geometry changes of the intake and exhaust will make for an even more accurate and representative model.

The software revealed that the peaks are due to intake and exhaust geometry. As the exhaust lengths are changed the results from the model show that the power and torque curves are dramatically altered when a short, medium or long piping of the same diameter is used. The short exhaust pipe examined is 20 inches, medium is 30 inches, and the long pipe is 50 inches to show the extremes of the three situations. The short pipe peaked at high speeds and had a higher peak power, but the peak was past the rev-limit of the engine. The medium-length pipe has a high peak at 8000 rpm but there is a loss of power at 5000 rpm because the characteristics of that geometry give resonance peaks before and after that point. The long pipe peaked at 8000 rpm and had a smooth increase in power with no local peaks but power dropped off after 8000 rpm much more than the other setups. The optimal setup will smooth out the power and torque curves at lower engine revolutions while maintaining or increasing peak power. Thus, the area under the curve increases and improves overall performance and drivability.

Table 2: Percent Gain in Power and Torque with Varying Compression Ratio

	Percent of Stock	
Compression	Peak Torque	Peak Power
11.3	96.43	98.25
12.3	100.00	100.00
13.3	102.38	101.75

Since a new piston and camshafts are being utilized, a simulation of the new cams and raised compression ratio was run in WAVE. The compression ratios tested range from 11.3:1 to 13.3:1, where the stock value is a 12.3:1 compression ratio. The values tested are one point lower and one point higher than stock in order to find how sensitive power gains are to changing the compression in this range. The results from the simulation are provided in Table 2, and show that as compression increases the incremental gain is decreased. Therefore, by adding a piston which increases compression to 13.5:1 will likely net less than a two percent improvement in power.

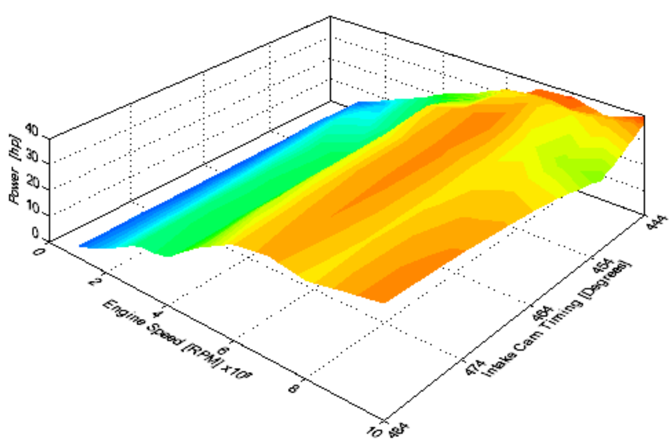


Figure 7: Intake Cam Timing Sweep Keeping Stock Exhaust Timing

The simulation of cam timing effects includes a sweep of both intake and exhaust cam timing starting at stock, and then changing timing by 20 degrees retarded and advanced in degrees of crankshaft rotation. The results from WAVE show that best timing for the intake cam is about ten degrees advanced from stock (Figure 7) and the exhaust cam makes best power at ten to fifteen degrees advanced. Both sweeps show that there is a fairly wide range of cam timing where power is not affected much. Therefore, WAVE predicts that power is not sensitive to intake cam timing in the range of five degrees retarded to ten degrees advanced from stock. Exhaust cam timing does not really affect power from five degrees retarded to twenty degrees advanced.

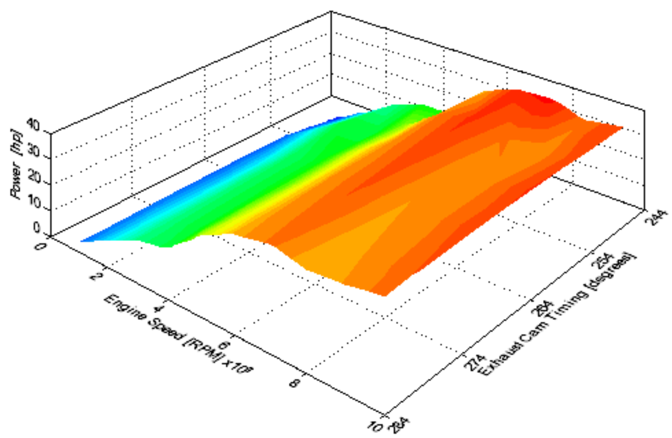


Figure 8: Exhaust Cam Timing Sweep Keeping Stock Intake Timing

This result is intuitive because intake valve timing is more important than exhaust timing. The intake valve closure point determines maximum cylinder pressure which is related to the engine's dynamic compression ratio. Since the intake valve closes after the piston reaches bottom dead center and starts moving up, the mixture actually gets compressed less than the static compression ratio (12.3:1 stock) dictates. The later the intake valve closes, the more maximum cylinder pressure decreases, thus compression is bled off by the intake valve. Therefore, dynamic compression refers to the actual amount that the mixture is compressed, factoring in the intake valve events. Exhaust cam timing is not as critical as intake timing, as shown in Figure 8, because the opening of the valve releases mostly burned gasses. The closure point of the exhaust valve and the opening of the intake valve are part of valve overlap where intake and exhaust valves are open simultaneously, affecting cylinder scavenging. Cylinder scavenging is the ability to fill and empty the cylinder effectively. Thus, the exhaust valve affects scavenging and release of cylinder pressure, but both of these events are not as important as the intake closure point. These effects are displayed in the results of the timing sweep, where engine power is less dependent on exhaust valve timing than intake valve timing.

When using the best timing of intake and exhaust cams combined, the model predicts peak power to be 44 hp and peak torque to be 33 lb-ft.

6.2 Gearing

Gearing for the Cal Poly FormulaSAE car is essential. By rules, the team is not allowed to make any changes to the vehicle once it has passed technical inspection. Selecting the correct gearing is one of the easiest and simple adjustments that can be made on the car and by selecting the correct gear ratios we can strike the best compromise between pure acceleration and autocross gearing to give the team the best overall performing and scoring car. In 2008 the team used a lap simulation to determine that using a 40 tooth sprocket in 1st-3rd gear was the fastest of the investigated gear ratios. There are significantly more gearing possibilities than were investigated and this analysis determines what gearing ratios have the most potential for success.

6.2.1 Optimizing the Current Gear Ratios

The first step in gearing is to figure out what the car currently has. The most basic way to quantify the current cars ability to use its gears is to build a tractive effort curve. The tractive effort curve shows the maximum force which could be

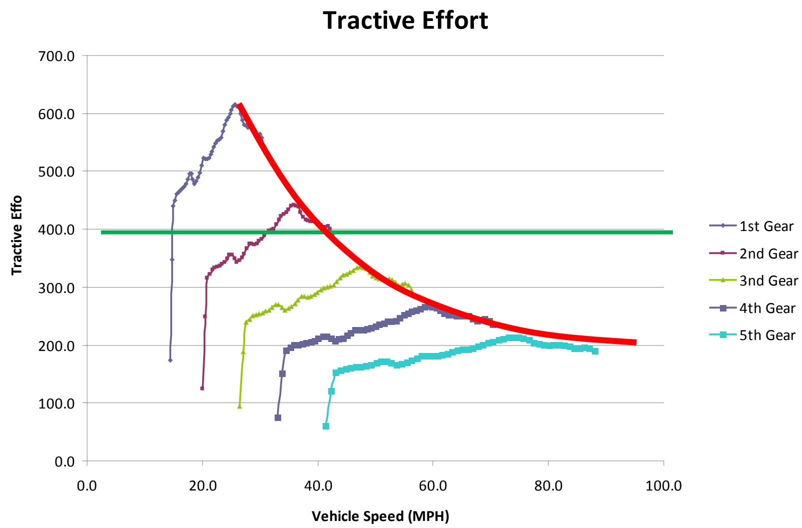


Figure 9: Baseline Tractive Effort Curve

produced by the wheels in any given gear, at any given vehicle speed. The basic tractive effort curve for the Cal Poly car can be seen in Figure 9. Both the velocity and force are determined for each gear using the gear ratio of the respective gear and engine speed. Vehicle velocity and tractive effort calculations are found using the equations B.1 and B.2 in Appendix B.

These calculations are then determined for each gear and each engine speed to create a full tractive effort curve. In a correctly geared vehicle the individual gear lines will connect. The point at which each gear curve intercepts next gear's curve represents the ideal engine speed or vehicle speed to change gears. In the approximate stock curve the gaps are present because the engine simply runs out of RPM. The 20mm restrictor limits the amount of air the engine can intake, which in practice limits the engine's speed to 10,000 RPM. This clearly shows that we either need to change our gearing and, or change our torque curve.

In addition there is a maximum tractive force that can be provided by the tire, green line. This can be found by multiplying the normal force on the tire by the coefficient of friction for the tire. To find this line, the FormulaSAE Suspension team uses

$$F_{Max} = 1.4N \text{ or } 390 \text{ lbs} \tag{6.1}$$

The final line is the maximum tractive effort line, red line. This is the tractive effort line which connects all the peak tractive effort points. Minimizing the area between this curve and each gears individual tractive effort curve results in more overall force produced by the wheels which in turn means faster acceleration.

Changing gearing will be difficult because the transmission is built into the engine. In order to change a gear the team would have to separate the engine casing then bond it back together after making the change. Because the previous process is time consuming and has a lot of potential for mistake, resulting in larger engine problems, we decided to look at a couple easy gearing changes and focus on developing a torque curve that is more suited to our car.

6.2.2 Using Non-Traditional Gears

The autocross and endurance events are the highest point paying events in the dynamic portion of the competition. As a result these two events are the ones that most teams need to design for. These courses are designed to have an average speed of around 35 mph and a top speed of about 60-65 mph. Because these courses are so tight there are a lot of shifts, most teams only use gears 1-3. In the WR450, these gears have large variations in gear ratios. For our car, the transmission ratios are per Table 3.

Table 3 : Table of Transmission Ratios

Gear	Ratio
1^{st}	2.417
2^{nd}	1.733
3^{rd}	1.313
4^{th}	1.05
5^{th}	0.84

For our car the gear ratios between 1^{st} and 2^{nd} and 2^{nd} and 3^{rd} are much greater than the gaps between 3^{rd} through 5^{th} gear. By changing the rear sprocket ratio we can make it so that 3^{rd} through 5^{th} gear can be used in the 15-65mph range.

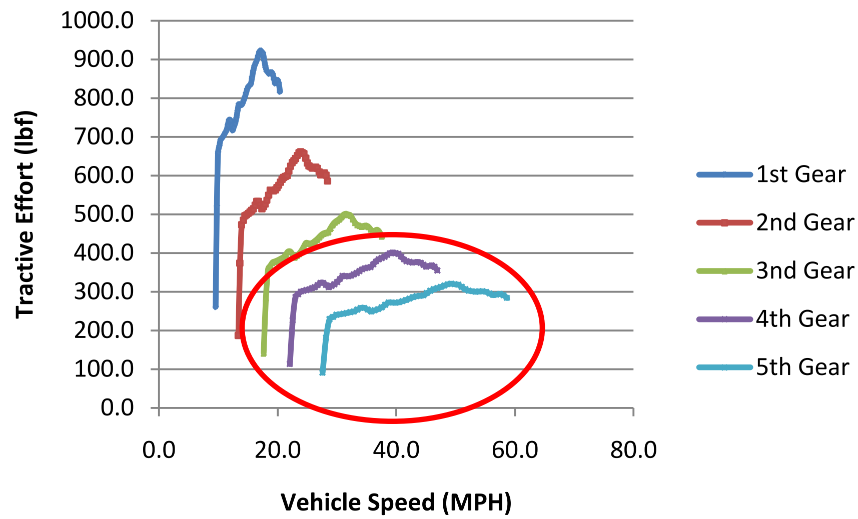


Figure 10: Tractive Effort Curve for Increased Sprocket

This change is shown in Figure 10. Note that making this change allows us to use the curves that are closer together. This means less of a drop in tractive force when shifting which could help with drivability of the vehicle.

To determine which gears we would use and what rear sprocket ratio is need, time to speed calculations were performed to determine which options have the most potential for success. (See Appendix A) If the acceleration time is not significantly increased then the change is not a useful option for the team.

Time to Speed is a way to simulate a vehicles acceleration time by using small increments of average acceleration. In this calculation the car can be seen as a simple mass system accelerating. The equation of motion is very simply $F = ma$. We don't know acceleration but based from the ground speed equation from tractive effort we do know vehicle velocity at each engine speed. We also know the force at the wheels for each engine speed. So, we can use the change in velocity over the change in time to represent acceleration. See equations A.3-A.5 Appendix A.

Using these equations we can determine the amount of time it will take to increase speed a small amount. Once all these smaller amounts have been determined we can sum them and add in shift times to determine total acceleration time. The acceleration times are then compared to determine which has the most potential for success. Table 4 shows that it is theoretically possible that using 2nd-4th gear will result in a significantly faster acceleration time than the current gearing.

Table 4: Time to Speed for Different Sprocket Ratios

Sprocket Ratio	Gears Used	Acceleration Time	0-60MPH Time	Top Speed (MPH)
40/14	1-3	4.50	3.57	101
48/14	2-4	4.08	3.45	84

In addition to acceleration time, drivability of the vehicle is very important. Based on the maximum available tractive force of the tire the car becomes traction limited in the first gear for every possible gear combination. An example of this can be seen in Figure 11. Having the ability to spin the tires at any given time requires more finesse from the driver making the car more difficult to drive. This gearing arrangement is designed to be driven in 3^{rd}-5^{th} gear. Based on the 390lb tractive force limit 1^{st}-3^{rd} gear are all traction limited at 390 lbs, seen by the plateau for each of those gears. For the autocross and endurance events most of the time spent in low gear is cornering and accelerating out of a corner. Ideally, to gain the most acceleration out of a corner we want to have the maximum amount of force at the minimum corner speed. This allows for the engine to produce the maximum amount of force possible for the longest period of time before requiring a shift. If the minimum corner speed is in the middle of the powerband for a gear, then the vehicle is not utilizing the maximum amount of acceleration it could. For the FormulaSAE competition on any given year, the minimum speed is 15mph. As a result, the base of the powerband, traction limited or not, would ideally begin at 15 mph.

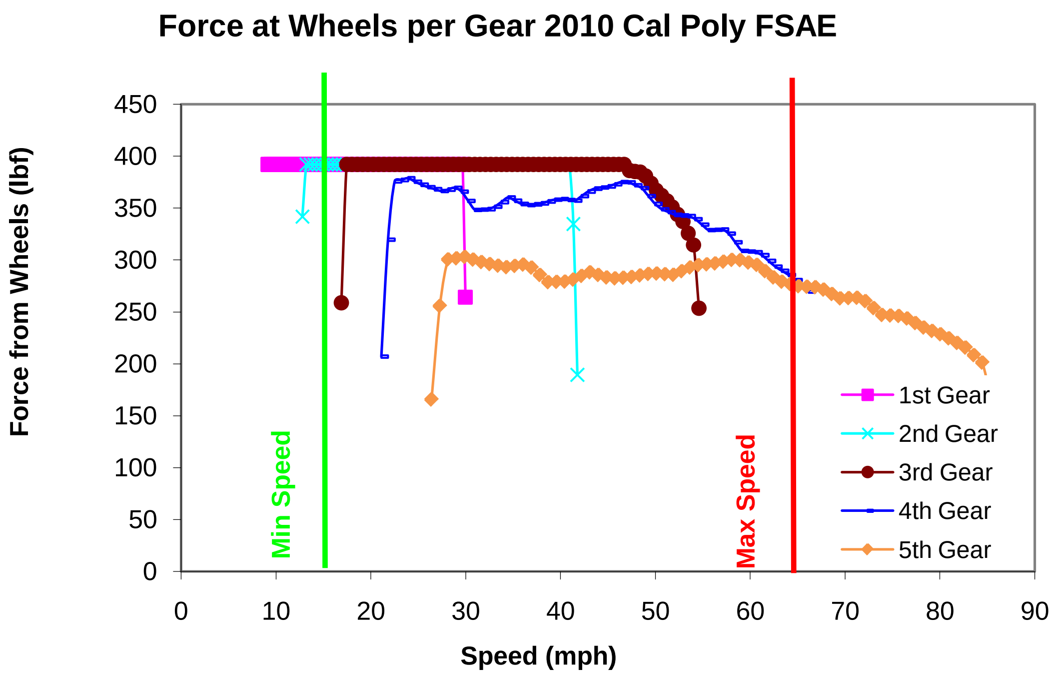


Figure 11: Tractive Effort of Different Gears

When accelerating out of a corner a progressive throttle input is necessary to prevent the car from under steering or throttle induced over steering. Thus the driver will not be giving the car full throttle and will not necessarily be asking the engine to produce more than the traction limit. It is however, advantageous to be able to reach the traction limit at any given time. If a good driver can stay on the edge of traction then theoretically they can achieve the maximum vehicle acceleration. To determine which set of gears, 1st-4th, or 3rd-5th is better for the competition, we will have to perform on car testing on a repeatable course.

6.2.3 Using Gears from other Yamaha 450 Models

To fully explore the extent of gearing possibilities we can input the YFZ Quad gear ratios into our models. The YFZ Quad engine has closer gear ratios than the WR because it requires more torque in each gear to power through resistive terrain and does not need the top speed capabilities that the motorcycle has. The gear ratios are per Table 5,

Table 5: Table of Transmission Ratios for YFZ450 Quad

Gear	Ratio
1st	2.416
2nd	1.928
3rd	1.562
4th	1.277
5th	1.050

To create a legitimate comparison the two set of gear ratios were compared with Time to Speed calculations. Both sets of gear ratios used the same power curve, primary reduction ratio (2.818) and final drive (sprocket) ratio for the initial comparison. When comparing 3rd-5th gear times, the YFZ Quad ratios required a smaller rear sprocket to achieve the necessary top speed. The results are summarized in Table 6.

Table 6: Time to Speed for WR and YFZ Gears

Engine	**Gears Used**	**Sprocket Ratio**	**Min Speed (MPH)**	**Max Speed (MPH)**	**Acceleration Time(s)/# Shifts**	**0-60MPH Time(s)/# Shifts**
WR 450	1-4	40/14	14.37	101.17	4.50/3	3.57/2
YFZ 450	1-4	40/14	14.38	80.94	4.53/3	3,82/3

From this analysis it appears that the WR 450 gearing, will provide quicker acceleration times than using the YFZ gears. Using the YFZ 450 gears seems to be an unnecessary risk. Although they maintain a competitive acceleration time, the times are slower and the added risk associated with changing the gears is not worth taking the time to test.

6.2.4 Final Drive Ratio

Once the transmission has been selected, the next step in the finalizing the powertrain is selecting the final drive ratio. In the past the FormulaSAE team has used Time to Speed calculations to pick sprocket size. Last year they used their lap simulation to confirm the result and actually found that 40 tooth sprocket was fastest for autocross and endurance. Again using the stock power curve as measured from the WR 450 bike, 3 different sprocket ratios were analyzed using 1st-4th gears. (Table 7)

Table 7: Time to Speed for Different Final Drive Ratios

Sprocket Ratio	Gears Used	Acceleration Time	0-60MPH Time	Top Speed (MPH)
33/14	1-3	4.14	3.44	122
40/14	1-3	4.5	3.57	101
48/14	1-4	4.59	3.96	84

From this analysis it appears that the 33 tooth sprocket will be a much better choice to maximize acceleration in all situations. In these three gears it becomes apparent that the design is a tradeoff between maximizing force available to the wheels and having the ability to always exceed that force. With the 33 tooth rear sprocket the car is traction limited in 1st gear only but is not pushing the tires to its maximum in 2nd. However, the vehicle is reaching a higher speed in each gear with the smaller sprocket thus minimizing the number of shifts. With the 40 and 48 tooth sprockets the vehicle is traction limited in 1st and 2nd gear, while having a lower speed capability in both. This trade off ultimately comes down to driver preference and ability. If the driver can handle the extra power and use it to their advantage, then the larger sprocket maybe faster. If the driver is a novice, being able to put the pedal to the floor without the car spinning provide an ease of driving and faster times.

Table 8: Comparison of 3 Best Gearing Options

Sprocket Ratio	Gears Used	Acceleration Time	0-60MPH Time	Top Speed (MPH)
33/14	1-3	4.14	3.44	122
40/14	1-3	4.50	3.57	101
48/14	2-4	4.08	3.45	84
52/14	2-5	4.42	3.41	78

Table 8 shows the vehicle's current set up, 40/14 using gears 1-3 and compares it with the three best options. From this analysis the designs with the most potential are using a 48 tooth rear sprocket while using 2nd-4th gear and a 33 tooth sprocket while using 1st-3rd gear. These arrangements provide the fastest acceleration time with a comparable 0-60 time. The final step in this process is to test. What we would like to do is run both acceleration and autocross test runs back to back. If the designed arrangement yields a faster time than the current car then that will be used, verifying our design.

6.3 Valve Timing

The ultimate goal of the camshaft design is to create a torque curve that is more suited to our car's operating range. The current camshaft is designed for the stock WR450 endurance motorcycle. This motorcycle is designed to run up to 12000 rpm as seen by the stock dyno curve

(Figure 12) and has a very broad torque curve between 6500-10500 RPM, about 2000 RPM higher than we'd like to have it.

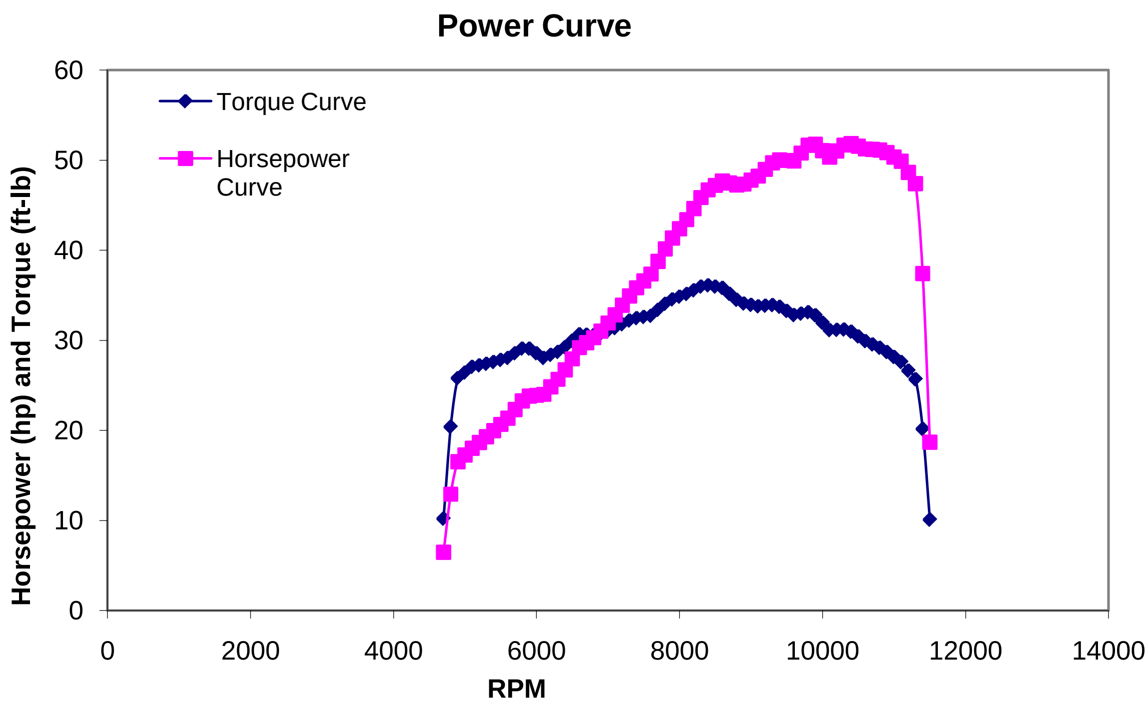


Figure 12: Power Curve of Stock WR450 Engine

The high peak of the torque curve occurs around 8300 rpm with a horsepower peak at 10,400 rpm. At this engine speed the amount of air reaching the engine is significantly reduced due to the restrictor. Grinding a cam is a process that we cannot accurately perform here at Cal Poly. With that in mind the design plan is to design a cam shaft for our engine, as if it did not have one. Once all the specifications are established we will either purchase an aftermarket cam or have one custom ground.

6.3.1 Tractive Effort Curve Modification

The first step in determining a torque curve is to close the gaps in the tractive effort curve. The current tractive effort curve (Figure 13: Stock Tractive Effort) has gaps, which represent a large drop in tractive force when shifting. These drops in tractive force are undesirable in a vehicle.

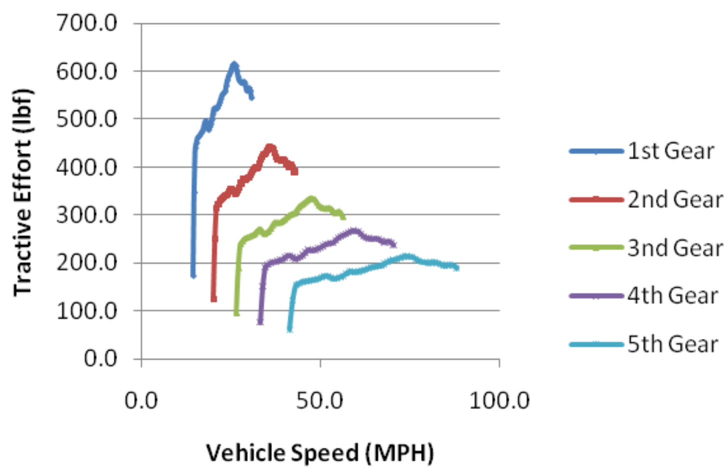


Figure 13: Stock Tractive Effort

To determine what general type of curve meet our requirements we input extreme torque curves to our tractive effort spreadsheet. We ensured that the peak torque in each curve was the same so our focus was solely on the shape of the curve

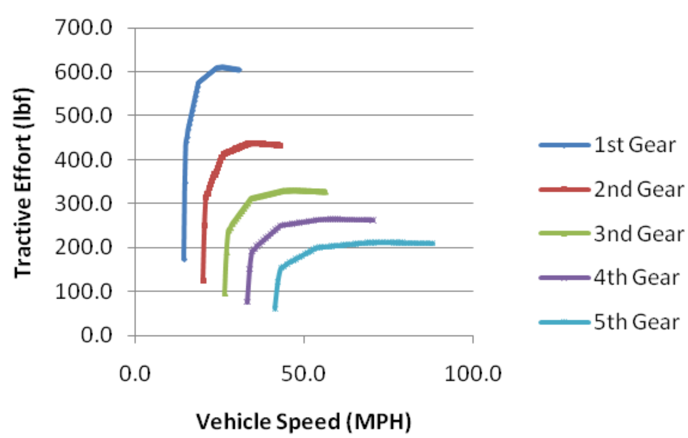


Figure 14: Flat Torque Curve Tractive Effort

A flat torque curve (Figure 14) is often described as “ideal” by enthusiasts, but this opinion is usually unsupported. This curve results in a tractive effort curve that does not peak but does not drop off either. Not having a drop in tractive force is good, but it makes the drop in force between gears even more pronounced than the original curve, which is undesirable. We then created a curve that peaked early, around 5000 rpm and then dropped off (Figure 15). As we expected the tractive curve peaked then dropped down to intersect with the curve from the next gear. This curve, although much more drastic than we would like is more of the curve we would like to have.

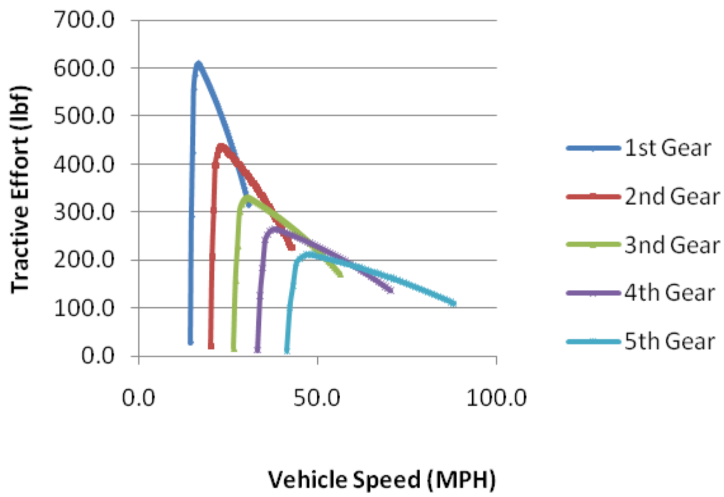


Figure 15: Early Torque Peak Tractive Effort Curve

Finally for sake of experiment we created a curve that peaked at the end of the rpm range. This curve also created a large gap like the flat curve but did not have an early peak, instead it peaked late creating the worst tractive effort curve of the group. (Figure 16)

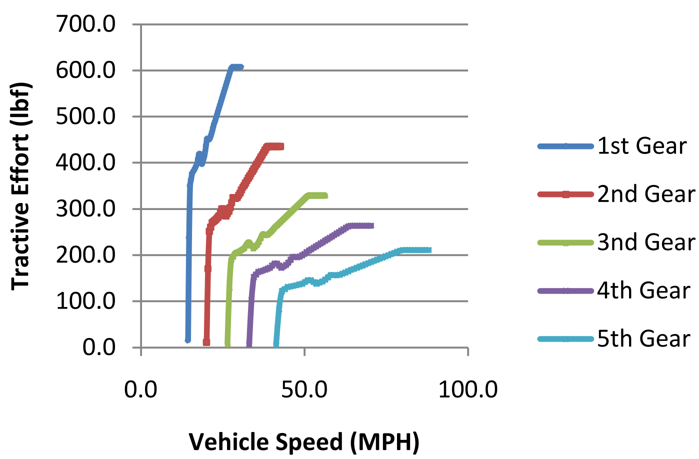


Figure 16: Rear Torque Peak Tractive Effort Curve.

For the FormulaSAE car a powerband that peaks earlier in the rpm range and maintains its torque rating towards the end of the rpm range is more effective. Ideally, toward the end of the rpm range the torque would begin to drop off, thus preventing the sudden drop in tractive effort between gears. The current powerband begins at 4700rpm and ends at 12000rpm. If we were to maintain the current length of the powerband, the new curve would begin around 3500rpm and end around 10,000rpm, or the maximum engine speed at which the restrictor can operate without significant restriction of flow.

6.3.2 Determining Maximum Valve Lift

In an ideal intake the intake valve opening(s) will be the point of maximum restriction of the intake. With that said, designing a valve so that it does not choke at maximum flow is critical for an engine to perform well at higher speeds. This is where design of the valve lift will start.

For this project we will retain the stock valves from the WR450 head. It currently has three intake valves each with a diameter of 27mm and two exhaust valves with a diameter of 28mm. The first step in the design is picking a Mach index (Z) to design for. The mach index is a ratio of inlet velocity to sonic velocity. Standard practice is to keep the Mach index below 0.6. This is justified in a plot of Volumetric Efficiency for various Mach indexes. (Figure 17) From this plot it is easy to see that any Mach index above 0.6 will result in lower than ideal volumetric efficiency.

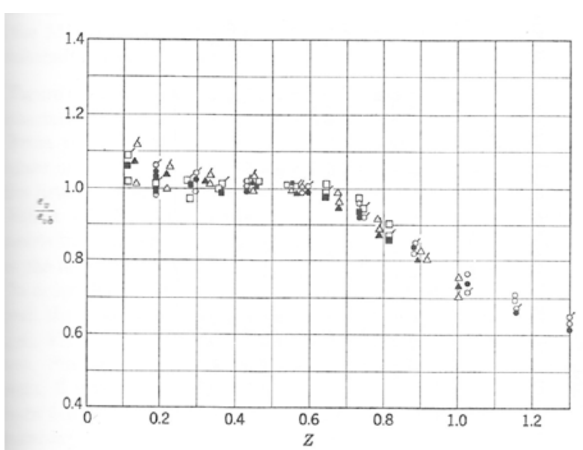


Figure 17: Effects of Mach index (x-axis) on Volumetric Efficiency (y-axis)

A Mach index less than 0.6 provides extra assurance that the flow will not be sonic through the valve but will require a larger flow area resulting, in our case, in extra lift. Once the Mach index has been determined, design can begin.

When designing the valve the end result is a valve effective area. This is the maximum area through which air will flow when the valve is open. for low lift situations the effective area is the area of the valve or[2],

$$A_{f} = C_f A_v = C_f \frac{\pi}{4} d^2 \tag{6.2}$$

where d is the valve diameter. At higher lifts however, the area next to the valve is larger so the effective area becomes[2],

$$A_f = C_d A_v = C_d \pi d l \tag{6.3}$$

where l is the valve lift and C_f and C_d are flow coefficients. We are designing our maximum lift, thus if we can determine the effective area needed at our maximum engine speed then we can determine what the maximum valve lift needs to be. Beginning with the intake valve, to determine the maximum effective area the following equation is used[2],

$$Z = \frac{\frac{\pi}{4} b^2}{\bar{A}_f} \frac{\bar{U}_p}{c_i} \tag{6.4}$$

This equation provides the mean effective area based on the Mach index, bore of the piston, b, the mean piston speed, $\bar{U}_p$, and the speed of sound in the intake tract, c_i. For a full calculation see Appendix C.2. Once the effective area is calculated it must be divided by an average flow coefficient to determine the representative area of the valve. Once the valve representative area is calculated, the lift can be calculated with equation (6.3).

The exhaust can be calculated in the same way but can be simplified down into an equation relating effective area of both the intake and exhaust valves to a ratio gas temperatures[2].

$$\frac{\bar{A}_e}{\bar{A}_i} = \left(\frac{T_i}{T_e}\right)^{1/2} \tag{6.5}$$

This equation assumes a Mach index of 0.6 through the exhaust and intake valve. In addition to this equation it is noted that standard practice sizes the exhaust area between 70-80% of the intake area[1]. Both calculations were performed and it was found that assuming an exhaust area of $A_e=0.7A_i$ required a larger amount of lift. Table 9 provides the results of the calculations on maximum necessary lift for the intake and exhaust valves (AppendixC.2).

Table 9: Valve Lift Needed for Different Engine Speeds

Engine speed (rpm)	Intake Valve Lift (in)	Exhaust Valve Lift (in)
10000	0.3	0.3
11000	0.33	0.33
12000	0.36	0.36

Per our intake design the restrictor will likely restrict engine speed to around 10,000rpm. As a result the team will probably set the ECU to limit the engine speed to 10,000rpm. In order to ensure that the valve lift is not the component limiting engine performance and maximum engine speed, the valve lift must be designed to choke *after* the restrictor would. To get the most performance, our valvetrain will be designed to run at 11,000 RPM, a higher RPM than the engine will ever see. So, our designed cam lift will be between around 0.33 inches.

6.3.3 Valve Duration and Timing

There is no explicit way to design the valve duration and timing. Many engine tuners have compiled their experience grinding and testing custom cams to come up with a "standard'' or trends for most engines. Due to this large level of uncertainty and the valve duration and timing's dependence on a specific application, the WAVE Model will be essential in the design of cam duration and timing. We cannot test hundreds of variations so we will be testing a select few that will give us an understanding of what we want in our final design.

Internal Combustion Engine by Ferguson and Kirkpatrick recommends the following for a high performance camshaft:

Table 10: Recommended Parameters for a High-Performance Cam, from Ferguson & Kirkpatrick

Valve	Open	Close	Duration
Intake	30^{o} Before TDC	75^{o} After BDC	285^{o}
Exhaust	70^{o} Before BDC	35^{o} After TDC	285^{o}

These values are not given at any particular value of lift, typically given at 0.040-0.050'', making it difficult to pick a design based on these numbers. Standard practice states that a longer duration cam is better for high rpm as the valves are open for a longer period of time allowing for more flow from kinetic effects. However, low end performance and idle suffer from a long duration because the valve is opened for such a long period of time that a large amount of cylinder pressure is lost in the compression stroke. For the FormulaSAE car we will try to design a cam that will perform well in the 8,000-10,000rpm range. We plan to gain back the potentially lost power from 5,000-8,000 rpm through intake and exhaust resonance tuning. The current cam shaft has a duration of 250^{o} at 0.040'' of lift. This seems very moderate for a high revving engine. In WAVE we will test this cam, against a Hot Cams aftermarket cam.

In addition to the total duration, when each valve opens and closes is critical for the performance of the engine. For a high performance, higher revving engine having a significant overlap between when the intake valve opens and when the exhaust valve closes is critical. This overlap provides the scavenging effects where exhaust gas pressure and momentum draws in additional intake air. It is a common practice to advance the cam to gain some mid range horsepower. By advancing the cam, the intake valve is opened earlier allowing the intake charge in sooner without as much flow leaving due to scavenging. Unfortunately too much advance will reduce scavenging too much and there won't be enough flow at high rpm to make max power. Although typically accepted as an easy way to gain power we will run a couple variations through WAVE to evaluate which solution is best for our application. In WAVE we will run the following models,

1) Current Cam Shaft from the car.
2) Advance 4^{o}
3) Advance 8^{o}
4) Retard 4^{o}

By performing these for simple approximations we should be able to quantify how much power and torque there is to gain and where in the RPM range it can be gained.

Figure 18: Falicon Cam Sprocket Set

The current intake valve timing corresponds with the recommendations from Ferguson and Kirkpatrick, the exhaust valve is advanced by almost 45^{o}. This will likely be a source for a large gain from the cam timing. This is, however the timing that Yamaha has installed stock. In order to change this timing and tweak the future timing we will purchase or manufacture a cam timing adjustment gear such as the gear made by Falicon (Figure 18: Falicon Cam Sprocket Set). These gears will allow for a significant and simple change in timing so we can fine tune our camshaft design.

6.3.5 Valve Train Dynamics

Valve train dynamics are critical to the life of the valve train as well as the engine. A valve train that is designed with no consideration of the dynamics will likely have high acceleration which will result in extremely large forces on the valves, buckets and cams. A small increase in acceleration at high engine speeds will result in a dramatic decrease in valve train life. The valve profile was measure with a dial indicator and degree wheel as seen in Figure 19.

Figure 19: Valve lift measurements

A full results table can be found in Appendix B. From these measurements we can derive the valve Velocity, Acceleration and Jerk. The lift profile looks very reasonable and smooth. When we begin to look at the velocity, but especially the acceleration curve we begin to see how very small, almost unnoticable changes in lift profle result in very large variations in acceleration and jerk which in turn result in very large changes in force, dramatically decreasing the life of the drive train. Typical valve dynamic curvescan be seen below[1]. (Figure 20)

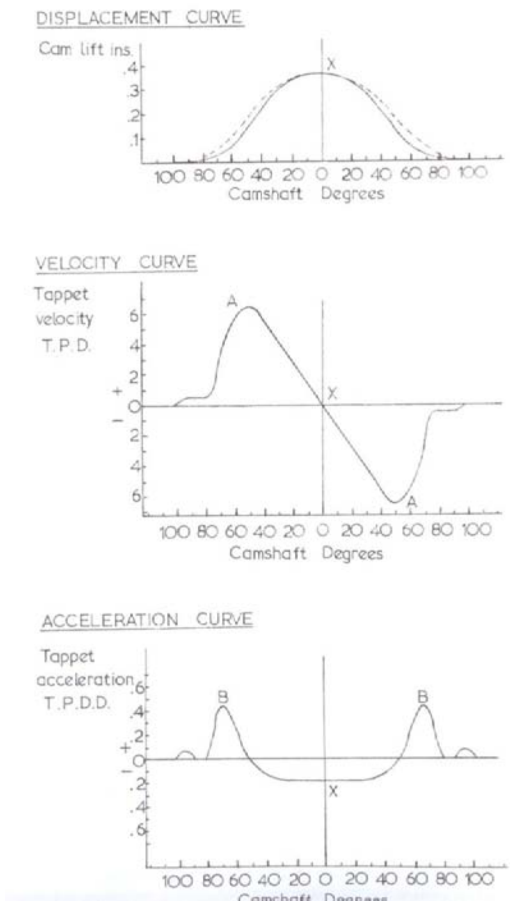


Figure 20: Typical Race Engine Valve Dynamics. Velocity and Acceleration units in Thousands of inches

These curves are all very smooth with no oscillation. Smooth curves, espcially acceleration, will result in a much smaller force on the valve train and a thus a longer-lasting system. The measured lash in our system is a result of measurement error and will need to be smoothed out for our valve train dynamics. The small variation is proof that cams are manufactured with very special and highly accurate machines.

6.3.6 Camshaft Options

Once we determine the final cam duration desired there are three options for manufacturing,

1) Use the stock cam with different timing
2) Purchase an aftermarket cam with desired lift and duration

3) Send out for a custom ground cam per our specifications

From this non WAVE verified analysis the aftermarket cam looks to be the most viable option. The current stock cam shaft has the following specifications.

Table 11: Current Cam Specifications

Valve	Lift (in)	Duration @ 0.040'' Lift
Intake	0.34	250^{o}
Exhaust	0.333	250^{o}

This cam is on the lower limits of our design and does not offer as much assurance for the assumptions we made about our inlet flow. Hot Cams, Inc. produces an aftermarket cam shaft set for our engine with the following specifications,

Table 12: Hot Cams, Inc. Cam Specifications

Valve	Lift (in)	Duration @ 0.040'' Lift
Intake	0.349	269^{o}
Exhaust	0.341	265^{o}

Figure 21: Aftermarket Cams, Hot Cams, Inc.

Based on the preliminary design this cam will give us enough lift to run our engine past 11,000 rpm and will provide a longer duration than the stock camshaft, putting us closer to the high performance cam shaft limit. Using an aftermarket cam also ensures that we do not have extreme valvetrain dynamic issues which could occur in a one-off cam.

6.4 Intake

6.4.1 Intake Tuning

The end goal with the Intake project is to provide more volumetric efficiency and improve engine breathing. This will in turn provide more torque and more power. The two goals of any intake are minimal restriction to flow and tuned resonance. Minimizing restriction to flow is simple enough; minimize bends, sudden expansions and contractions, and make the piping diameter sufficiently large. Tuned resonance is somewhat more complex.

Because the 4-stroke power cycle isn't steady-state, harmonics can make or break the breathing process of an engine. When the intake valve opens, it creates a rarefaction wave that propagates through the intake tract until it reaches a discontinuity (in Figure 22, a megaphone). Like any other wave, a discontinuity in the medium causes some degree of reflection. The pressure wave that is inverted as it is reflected and propagates back toward the intake valve as a compression wave. (Figure 22).

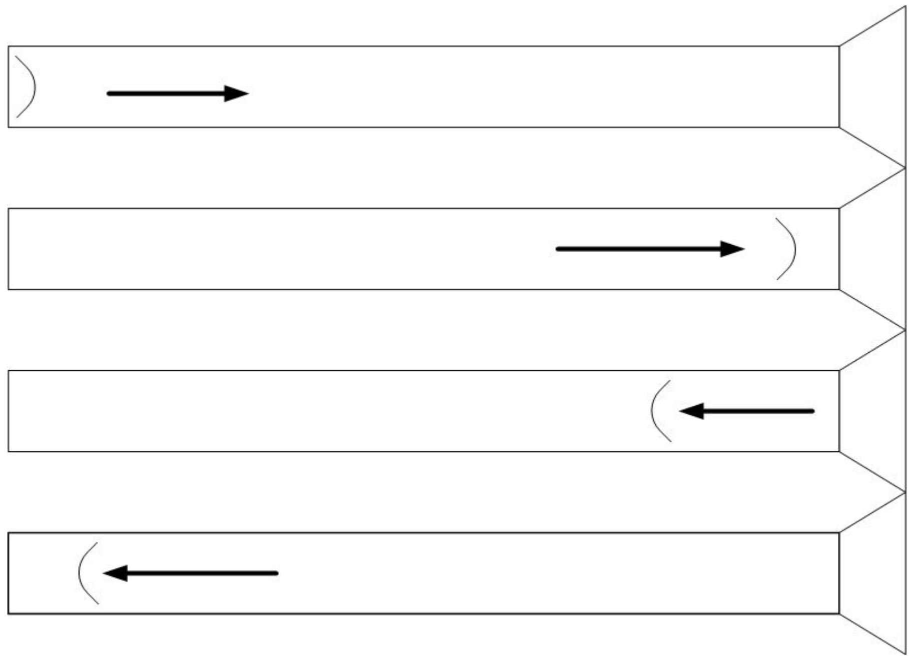


Figure 22: Pressure Wave in Idealized Intake

When the pipes are in resonance, the returning compression wave will arrive back at the intake valve just prior to its closing. This increases the pressure that the intake port sees, potentially to above atmospheric pressure. This causes a corresponding increase in engine volumetric efficiency, which manifests as torque.

The natural resonance of a given set of intake pipes will be at one particular frequency which will correspond to one particular engine speed. Shortening or lengthening the intake pipes can make this natural frequency higher or lower, respectively, which will result in more torque at higher or lower engine speeds, again respectively.

The current intake pipes on the Cal Poly FormulaSAE car were not designed to take resonance effects into account. They were constructed using simpler criteria such as placing the air intake in the free stream (also referred to as a "cold-air intake"), ease of manufacturing and available space on the car.

Table 13: Intake Tracts for the Engine at Different Speeds, Speeds of Interest Highlighted

Engine		Intake Valve			Wave travel distance in Valve Closed Time	Intake Tract length for number of wave round trips					
Rpm	Frequency	Freq	Cycle Time	Closed Time		1	2	3	4	5	6
	(Hz)	(Hz)	(s)	(s)	(m)	(m)	(m)	(m)	(m)	(m)	(m)
4000	66.67	33.3	0.030	0.011	3.83	1.91	0.96	0.64	0.48	0.38	0.32
5000	83.33	41.7	0.024	0.009	3.06	1.53	0.77	0.51	0.38	0.31	0.26
6000	100	50.0	0.020	0.008	2.55	1.28	0.64	0.43	0.32	0.26	0.21
7000	116.67	58.3	0.017	0.006	2.19	1.09	0.55	0.36	0.27	0.22	0.18
8000	133.33	66.7	0.015	0.006	1.91	0.96	0.48	0.32	0.24	0.19	0.16

We planned on designing new intake pipes to resonate at a frequency in or around the middle of our desired powerband. We plan on evaluating and tuning the resonant frequency of the intake pipes by modeling them in Ricardo WAVE and steady-state dyno testing. For more detail on this, refer to 8.2.2 Testing Procedure.

We also looked into variable-geometry features, such as an adjustable length intake pipe to adjust the resonant frequency. The driver could either set the resonant frequency lower for acceleration and higher for top-end speed, or possibly even have the car dynamically adjust the resonance to increase volumetric efficiency across a large swath of the powerband. This variable length intake tract could also account for barometric conditions, which will change the optimal length somewhat for a particular engine speed.

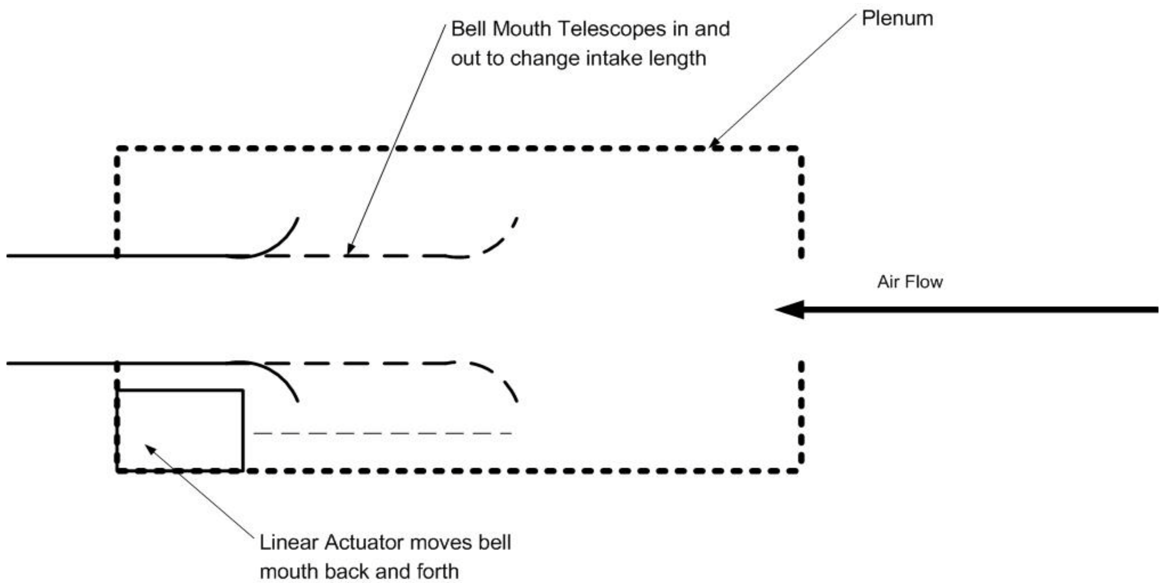


Figure 23: Proposed Design for Variable-Geometry Intake with Continuous Adjustment

Our first idea for variable geometry was to have a movable bell mouth at the end of the intake tract, just as the flow leaves the plenum (Figure 23). This bell mouth would move axially within a slightly larger pipe in order to change the overall length of the intake. This bell mouth would be moved by an actuator of some sort, and seek a position for the bell mouth that corresponds to a map of engine speed and barometric pressure. As the engine speed increased, the ECU would detect this rise and begin to shorten the intake tract.

We expect the engine to operate between 4,000 rpm to 8,000 rpm at competition. While the proposed variable length intake would ideally allow tuning along this entire range, the length would have to double (or halve) to allow this at one harmonic frequency. One solution to this would be to switch from one harmonic frequency to another during acceleration. This would allow the lowest and highest engine speeds to be in resonance, and also reduces greatly the stroke distance needed.

The current technical problem associated with this first design centered on the actuator itself. The actuator would need a stroke length of at least 150mm and the ability to traverse through that stroke length in 0.75 seconds. Most linear actuators are not capable of the combination of speed and stroke length this application will require, and those that do are prohibitively expensive and/or prohibitively large. We decided that the effort to locate or fabricate such an actuator would be excessive for the potential power gain.

We also explored several other conceptual configurations for a variable-geometry intake. These other configurations were discrete changes from one resonant length to another.

The first of these was to have a smaller Bell Mouth that could mate inside a larger one. The smaller bell mouth would tilt into and out of contact with the larger, stationary bell mouth. These could have a long or short configuration (see Figure 69: Tilting Bell Mouth Design, High Speed (short) Configuration and Figure 70: Tilting Bell Mouth Design, Low Speed (long) Configuration). This design could use a servomotor instead of a linear actuator, but we determined that the tolerance required for it to work properly would require too much of our limited time.

The second concept was to have both bell mouths stationary in the plenum, and have the ECU pick between them using a valve. We designed a test piece version (Figure 71), and also a one-piece rapid-prototyped final version (Figure 72) . We did not explore these designs due to our concerns about not being able to fully seal the valves, both at the intake tract to plenum interface and the intake tract to the atmosphere

.

6.4.2 Working around the Restrictor

The 20mm Restrictor is one of the thorniest problems we face. It greatly reduces the amount of air the engine can inhale, which provides its own problems of decreased air for combustion, and also creates additional problems as we attempt to work around it. The most common way to work around a restrictor is through the use of an intake plenum. The plenum, a chamber in series with the intake pipes, serves as a reservoir which can buffer out the intake pulse to a certain extent, which in turn allows the restrictor to see a constant flow rate lower than its theoretical peak flow rate with no plenum. Unfortunately, the plenum also buffers any pulse from the throttle body, which greatly reduces throttle response and thus drivability.

Preliminary calculations on the restrictor indicate that with an infinitely large plenum, the restrictor will reach sonic conditions at around 8,000 rpm. (Table 14 : Restrictor Choked Flow Calculations) With no plenum, and with the intake flow idealized as a square wave, the restrictor will reach sonic conditions at around 3,500 rpm, but will not be choked until 11,000 rpm. This illustrates the value of a plenum, but does little to show us what size will best balance throttle response with smooth flow. These calculations are summarized in Figure 24 below.

Table 14 : Restrictor Choked Flow Calculations

Ideal engine efflux				Ideal Plenum		No Plenum	
Rpm	Volume efflux		Air Velocity	Air Velocity	Mass efflux	Unsteady Air Vel	
	m^3/sec	cfm	m/sec	Percent of Mach	kg/sec	m/sec	% Mach
2000	0.0075	15.879	23.87	7%	0.0092	59.27	17%
3000	0.01125	23.819	35.81	11%	0.0138	88.91	26%
4000	0.015	31.759	47.75	14%	0.0184	118.54	35%
5000	0.01875	39.698	59.68	18%	0.0230	148.18	44%
6000	0.0225	47.638	71.62	21%	0.0276	177.81	52%
7000	0.02625	55.578	83.56	25%	0.0322	207.45	61%
8000	0.03	63.518	95.49	28%	0.0368	237.09	70%
9000	0.03375	71.457	107.43	32%	0.0413	266.72	78%
10000	0.0375	79.397	119.37	35%	0.0459	296.36	87%
11000	0.04125	87.337	131.30	39%	0.0505	325.99	96%
12000	0.045	95.276	143.24	42%	0.0551	355.63	105%

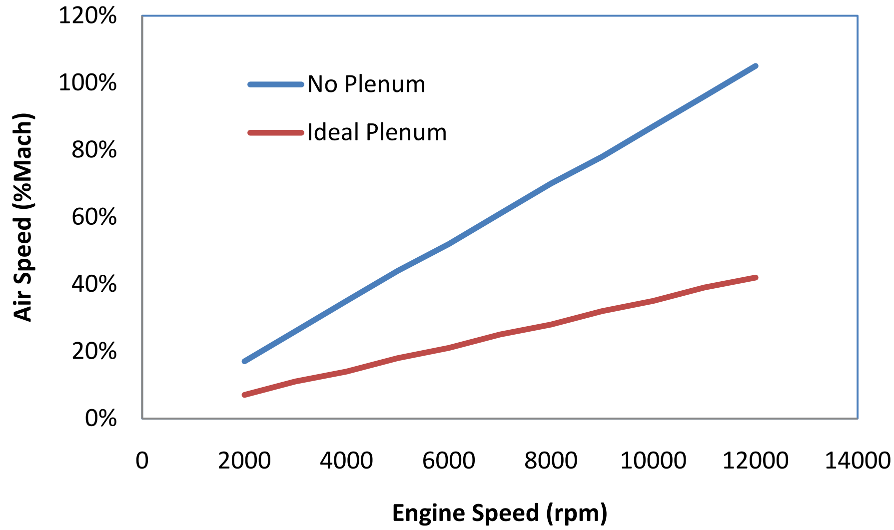


Figure 24: Effects of Plenum on Theoretical Air Flow speed at Restrictor

The Formula team's 2009 restrictor (Figure 68) is a venturi shape, and should already minimize head loss and Vena Contracta effects. Accordingly, we did not attempt any modification to this profile. We planned to evaluate the effect of plenum size and shape on the flow across the restrictor, in order to perhaps decrease the size of the plenum and thus the delay in throttle response. We designed a test plenum which allowed us to change its volume (Figure 25).

Figure 25: Picture of Test Plenum with Medium body and Short bell mouth

6.4.3 Bell Mouths

Bell Mouths, also called “Intake Trumpets”, serve to reduce the Vena Contracta effect, which occurs when fluid enters a pipe. Vena Contracta (see Figure 26) occurs at pipe entrances when fluid flow enters a pipe from a larger pipe or plenum. The fluid’s inertia causes the boundary layer to separate from the wall of the pipe, creating a ring of relatively stationary fluid. This reduces the effective diameter of the pipe entrance and causes head loss.

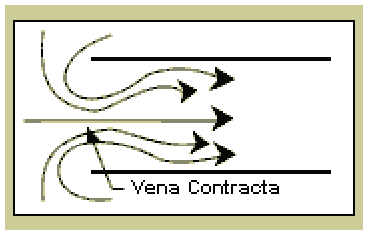


Figure 26: Vena Contracta Effect at Pipe Entrance

A Bell Mouth can reduce or even eliminate this effect. Figure 27 below shows how properly forming a Bell Mouth can eliminate almost all head loss associated with the pipe entrance.

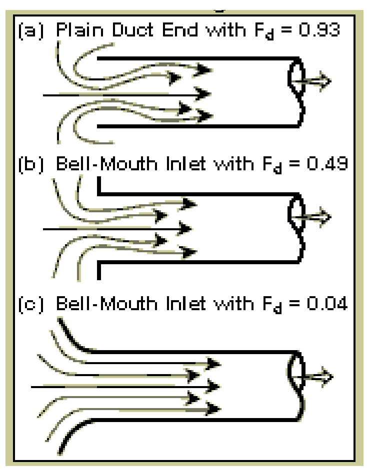


Figure 27: Comparison of Straight End with two Bell Mouth Shapes

We used Bell Mouths for the pipe exit from the Intake Plenum. We would also recommend the FormulaSAE team install a similar bell mouth at the throttle body. Our bell mouth follows an elliptical profile specified by Blair, et al. in an article published in *Race Engine Technology*. They specified an “ideal” dimension for an elliptical bell mouth, but also noted that the flow difference between their “ideal” and a non-ideal was only a percent or two. We used their profile for our bell mouth designs nonetheless. We designed a short and a long bell mouth, specified in Figure 73: Drawing of Long Bell Mouth and Figure 74: Drawing of Short Bell Mouth . The length of the bell mouth feature dictates the intensity of the resonant effect, as well as the range of its effects, longer bell mouths causing lower intensity effects over broader ranges. For a more detailed explanation of this phenomenon, see Figure 29 and its accompanying discussion on page 58.

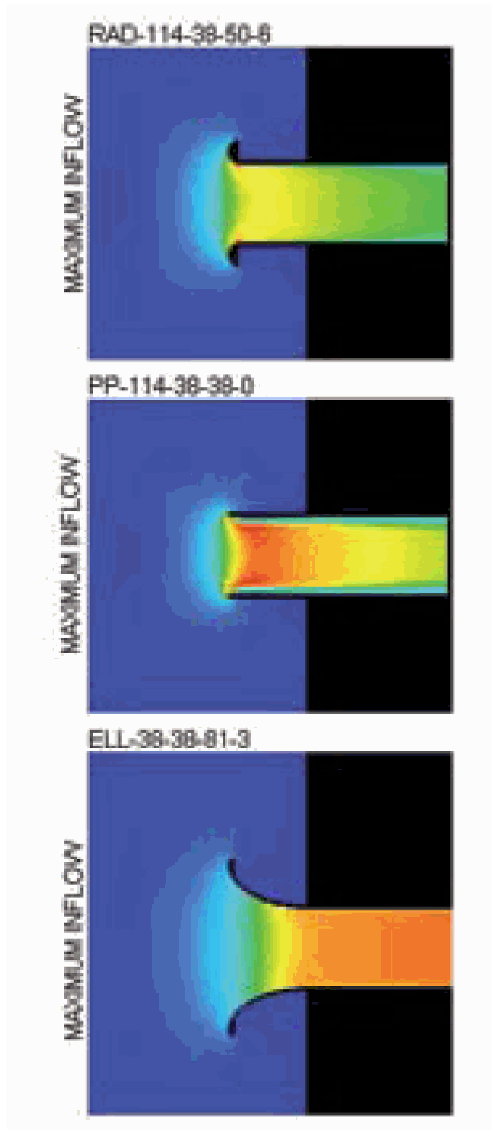


Figure 28: CFD Comparison of Different Bell Mouths, Elliptical at bottom. from Blair, et al.

6.5 Exhaust Tuning

The proposed plan for the exhaust is to improve scavenge and improve volumetric efficiency. We will utilize wave dynamics and tuned tract lengths to increase scavenge, and design a free-flowing exhaust system to reduce back pressure. Both of these will increase engine volumetric efficiency, which will in turn increase engine power and torque.

Wave dynamics affect the exhaust system the same way as they affect the intake. In the exhaust, we are trying to get exhaust gases out of the cylinder more completely (scavenging) to improve volumetric efficiency. As the exhaust valve opens, it creates a high pressure wave that travels down the exhaust tract. This wave is reflected back from the end of the pipe as a large expansion (low-pressure) wave. The goal is for the expansion wave to arrive at the exhaust port just before the exhaust valve closes. This under-pressure at the port can increase volumetric efficiency in the cylinder by pulling out more waste gases. This under-pressure can also help pull in fresh intake charge, but this effect is effect is dependent on overlap between intake and exhaust valves.

Table 15 below shows the different exhaust tract lengths for number of wave round trips. The calculations are similar to those for resonant intake lengths, the only difference being the speed of sound. In the exhaust, sonic speed increases due to much higher temperatures, exhaust resonant lengths are longer than intake lengths for a given engine speed and valve duration. At 1,600°F, the speed of sound is around 680 m/s.

Table 15: Exhaust Tract Lengths for the Single-Cylinder Engine at Various Speeds

Engine		Exhaust Valve			Wave travel distance in Valve Open Time	Exhaust Tract length for number of wave round trips					
Rpm	Freq	Freq	Cycle Time	Open Time		1	2	3	4	5	6
	(Hz)	(Hz)	(s)	(s)	(m)	(m)	(m)	(m)	(m)	(m)	(m)
4000	66.6	33.3	0.030	0.011	7.65	3.83	1.91	1.28	0.96	0.77	0.64
5000	83.3	41.7	0.024	0.009	6.12	3.06	1.53	1.02	0.77	0.61	0.51
6000	100	50.0	0.020	0.008	5.10	2.55	1.28	0.85	0.64	0.51	0.43
7000	116.6	58.3	0.017	0.006	4.37	2.19	1.09	0.73	0.55	0.44	0.36
8000	133.3	66.7	0.015	0.006	3.83	1.91	0.96	0.64	0.48	0.38	0.32

	Harmonic	1	2	3	4	5	6
5k Rpm	Tract Length (cm)	306.00	153.00	102.00	76.50	61.20	51.00
	Tract Length (in)	120.47	60.24	40.16	30.12	24.09	20.08
7k Rpm	Step Location (cm)	218.57	109.29	72.86	54.64	43.71	36.43
	Step Location (in)	86.05	43.03	28.68	21.51	17.21	14.34

	5k		7k		Stepped	
	14.55	header	14.55	header	14.55	header
	2.23	Intermediate pipe	5.10	Intermediate Pipe	3.00	Intermediate 7k
	3.88	step	3.88	step	0	step
					3.10	Intermediate 5k
	19.5	Muffler	19.5	Muffler	19.5	Muffler
Total Length	40.16		43.03		40.16	

A step affects pressure waves just as the end of the exhaust tract does; it's a sudden expansion (discontinuity) that causes a portion of the wave to be reflected. (Figure 29) Our step (Figure 88) essentially creates two resonant peaks, but each is lower in magnitude than a peak from a pipe with only one resonant length. We made an exhaust with a l 5,000 rpm resonant length and then we put a step at the 7,000 rpm resonant length (measured from the port) to get a small boost at that speed too. Initially we ran with long pipe but changed to shorter pipe to reduce head loss. Long pipes allow lower, and thus stronger, harmonics of the pressure waves, but also have more head loss. Also narrower pipes have stronger resonance effects, due to the greater fluid velocity for a given volumetric flow rate, but again cause more head loss. As we were not able to predict the gains from resonance with high accuracy, we did not experiment with overly long or narrow pipes.

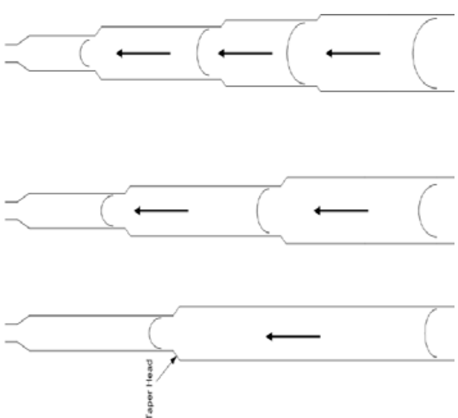


Figure 29: Example of a Stepped Exhaust Pipe

We also designed test sections for 5,000 and 7,000 rpm resonant lengths with no steps. All the exhaust configurations were adjustable by swapping out test sections. (See Figure 83 and Figure 84 for the two resonant length test sections)

Originally we designed the exhaust to have several steps but decided to buy a megaphone (Figure 82) instead. A megaphone is similar to multiple steps because in theory it reflects the wave at an infinite number of steps within a given resonance length (the start and end length of the megaphone). This creates a very broad resonance response, but its magnitude is lower than that of a single step. Flow separation occurs at expansions above 7 degrees, so our megaphone had an included angle less than 14°.

The new exhaust was developed out of mild steel tubing and an FMF straight through muffler. (Figure 30) Mild steel tubing was chosen because it is cheap, so it will help us stay inside our budget goals, and it is very easy to work with. The FMF Powerbomb 4 muffler was chosen because it was straight though, as desired, and it was the cheapest straight through muffler that could be purchased from any muffler manufacturer, approximately $240 new. The cost was a major factor as a good muffler from Yoshimura or even FMF can cost up to $500-$700.

Figure 30: FMF Power Bomb Muffler

6.6 Compression Ratio and Combustion Chamber

There are numerous methods of increasing the compression ratio of the engine. The three main methods that can be used are removing material from the head-block interface, adding material to the combustion chamber, or using a piston with a taller or domed top. The most reasonable method to raise compression in this case is installing a high compression piston.

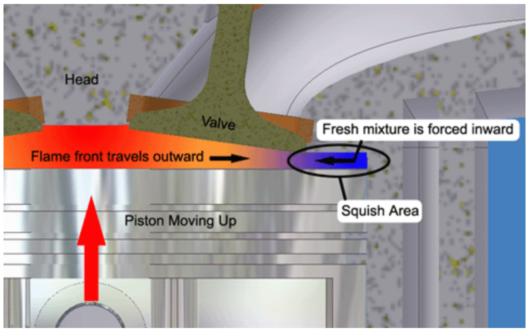


Figure 31: Squish effects. From meched.com

Using a domed or taller piston will promote squish within the combustion chamber. Squish refers to the volume of the air-fuel mixture that is "squished" by the piston, against the cylinder head. Squish is beneficial and increases if a piston is domed, since the piston profile comforms to the combustion chamber shape. A taller piston, or one with a different piston pin location, will increase squish by simply coming closer to the cylinder head at top-dead center (TDC). With the mixture being compressed in a smaller volume, the flame front created by the spark will have to travel less distance. This is beneficial because there is less of a chance the mixture will ignite before the spark occurs, often referred to as detonation, and a more complete burn of the fuel will be achieved by squishing the entire mixture into a smaller volume.

Figure 32: Quench area. The mixture gets squished where the combustion chamber is level with the head surface. From members.cox.net

There is a smaller chance of detonation because the quench area, referring to the area where the piston and cylinder head are almost touching, runs cooler (see Figure 31). With the heat trapped towards the center of the piston, the mixture is more uniform and hot spots that ignite the mixture prematurely are reduced. A more complete burn of the fuel occurs since the mixture is forced into a smaller area, gets more concentrated and thus more uniform.

Concerns of increasing compression include valve shrouding and the possibility of engine interference. Valve shrouding occurs when a surface within the combustion chamber is close enough to a valve such that when the valve opens, the flow of air and fuel is obstructed by the surface (i.e. cylinder head wall). Any alterations in combustion chamber shape must account for valve shrouding (see Figure 33).

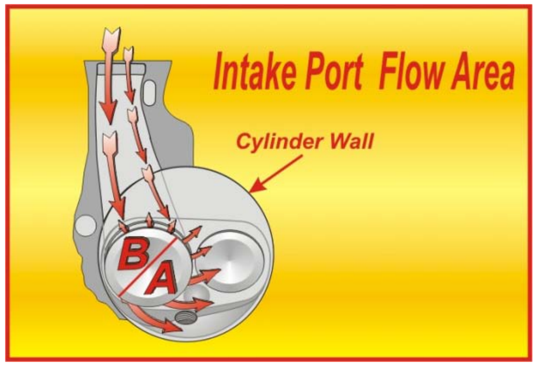


Figure 33: Valve Shrouding. Flow around area B is noticeably less than flow around area A because area B is shrouded. From gofastnews.com

A piston change does not have a noticeable effect on valve shrouding because the piston does not block the valve openings on an overhead cam (OHC) engine. If the compression is then raised by removing material from the block or head after installing the piston, then valve shrouding would be of concern.

Engine interference can arise from increasing valve lift and piston height. With a decreasing valve-to-piston clearance there is the increasing possibility of interference between the valve and the piston. The clearance will have to be measured to determine the maximum amount of lift that the valve can sustain before destroying itself and the piston. The intial method of measurement will be to place a deformable material on top of the piston, such as clay, and cycle the engine. The thickness of the material after turning the engine will deteremine how close the valves get to the piston. Accounting for thermal and kinetic effects, the maximum allowable lift will be the measured distance traveled minus 2mm.

JE Pistons makes a domed piston to increase compression to 13.5:1 and Venom Performance makes a piston to increase compression to 14:1. These pistons are available for the WR450 and can be easily purchased and installed. If it is found that the engine can sustain an even higher compression ratio on 100 octane fuel, then compression can be further increased by removing material from the cylinder head surface. The aforementioned considerations in this section will then dictate how much material can be removed.

7. Manufacturing

7.1 Intake

The intake has the largest number of components and was the most complex system to manufacture for our project. Consisting of multiple parts including the restrictor and multiple bell mouths made this our most machining and welding intensive system.

7.1.1 Restrictor

The restrictor is the most precise part of our design. The goal of the restrictor is to maximize the area in the restrictor while still meeting rules. To do this we attempt to machine the throat of the restrictor to as close to 20mm as possible. For the 2009 car the restrictor was machined out

Figure 35: Machining inlet portion of main restrictor section

Figure 34: Additional Restrictor diffuser length

of Delrin on a CNC lathe. The team found this to be very precise and the Delrin machined very well resulting in an extremely smooth part with a restrictor throat measuring 19.9mm. Because of the small geometry inside the restrictor, the length and size of the boring bar used to produce the inner profile was very limited. These tooling constraints required the restrictor to be cut in two portions. The first was the main part of the restrictor consisting of the inlet, throat and most of the diffuser. (Figure 35) By cutting the inlet and throat then flipping it to cut the diffuser, the boring bar only had to stick out a maximum of 2". This significantly reduced the amount of vibration resulting in a smooth part. The second part was the end of the diffuser adding an additional 1.5 inches to the maximum length achievable with the boring bar. (Figure 34) The final step is to bond the restrictor into a 1.5" aluminum tube with epoxy. This allows the restrictor to be rigidly mounted to another part of the intake without the risk of breaking it. This also provides a surface to weld or mount to.

7.1.2 Bell Mouths

There are two separate bell mouths used for intake testing. Both parts needed to be CNC machined on a lathe from 6061 aluminum. The third bell mouth design was deemed un-machinable as the elliptical shape proved to be difficult to machine due to vibration of the boring bar and the lack of a portion to clamp in the lathe chuck while machining the inner profile.

Figure 36: Body of bell mouth inner and outer profile

Each bell mouth starts as a length of bar stock 4" in diameter. Once clamped in the lathe on the "Bell" side, it was faced to length then drilled to 1" ID. This inner diameter will open the hole enough to use the largest boring bar in the Mustang '60 ME machine shop. Once the hole is drilled to 1'' the boring bar is hand fed the length of the part to open the hole to 1.25", or the minimum bore size for the boring bar. This step size was very large, 0.125'' off the radius and would likely either over load the machine or break either the tool or part if not fed very slowly by hand. The final inner diameter was then programmed into the lathe to ensure a constant feed and a smooth finish. The inner diameter was cut first because from machining experience, a thin walled part is less likely to vibrate if the inner diameter is performed first. This is primarily because turning tools are more rigid than boring bars as they only stick out of the tool holder a couple inches. The outer profile of the bell mouth was then cut. (Figure 36) At this point there is approximately 0.5" left as solid stock at the end of the part. This will become the opening of the bell mouth.

Figure 37: Completed Bell Mouth

To machine the opening of the bell mouth the part is taken out of the lathe and clamped by the long straight section just machined. Again a 1" hole was drilled followed by opening it up to 1.25" with the boring bar. This side is easier as it is shorter but still requires 2-3" of stick out. Once that hole is opened, the CNC program can be run to cut the opening of the bell mouth. (Figure 37)

Figure 38: Short Bell Mouth

Both bell mouths took approximately two hours total to code in CamWorks for the CNC. The long bell moth required approximately two hours of total machine time and the short bell mouth (Figure 38) required approximately an hour of total machine time. The elliptical bell mouth was attempted three different ways with each resulting in a thrown part or too much chatter to be useful.

7.1.3 Intake Runner and Plenum

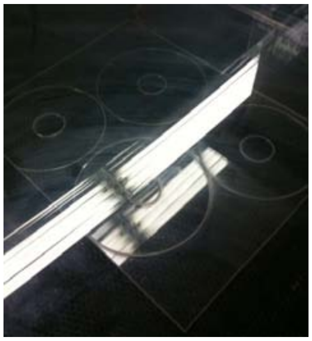


Figure 39: Plenum end caps on laser cutter

3 prototype plenums were manufactured for dyno testing. The current design calls for three acrylic plenums manufactured so they can easily be changed out. One 6" diameter tube was cut into three sections, with lengths of 5", 7" and 9". These sections are the three different plenum sizes. To cap these ends, a sheet of acrylic

was cut on the Mustang '60 laser cutter. (Figure 39) Each cap is two pieces bonded together to form a step that will fit snugly inside the plenum and seal the end. The Bell mouths were then welded to their respective runners which were then glued to the end caps. (Figure 41) Before being installed on the engine a hole for the manifold air temperature sensor needed to be tapped and a fuel injector mount had to be made.

Figure 42: Bell Mouths welded to runners

When designing the fuel injector mount the team wanted it to be easily removable, light weight and pointed as close to directly at the port as possible. After some time the injector mount in Figure 40 was developed. This design allows the fuel rail to be easily removed by pulling the pins holding it in place. This also prevents the injector from being over tightened as it has some freedom to move. In addition when fully welded the part is very rigid yet still lightweight. Both of these characteristics are very important on an engine that shakes as violently as ours does.

Figure 41: Injector Mount

When all these components are combined the following intake resulted. (Figure 42) Some parts

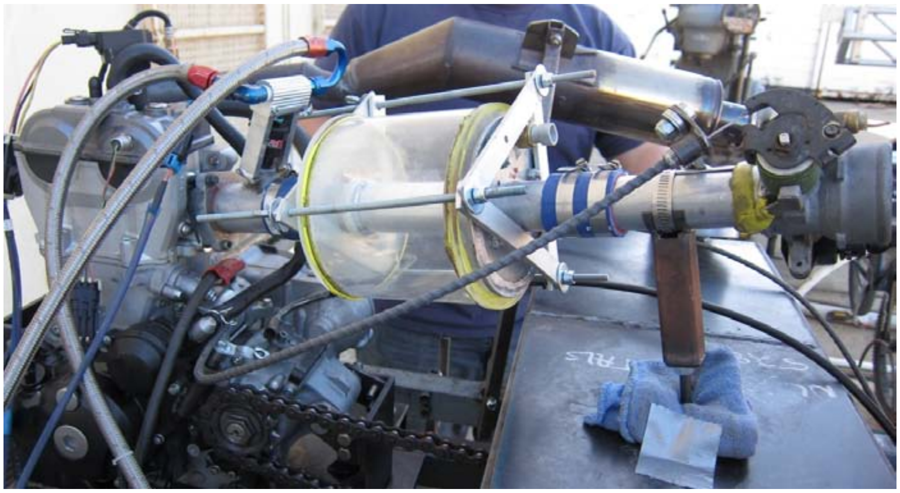


Figure 40: Fully assembled and operational SPEED intake

had to be modified so that they sealed and did not allow for vacuum leaks.

7.2 Exhaust

Figure 44: Exhaust tuned for 5000 RPM

The bulk of exhaust manufacturing consists of cutting and welding the tubing for the exhaust pipe. For the exhaust testing we decided to use mild steel tubing because of the much cheaper cost. A final exhaust should be built out of Stainless Steel because of its higher heat tolerance but much more expensive. From our exhaust calculations we had three different configurations to test; one tuned for 5,000 rpm, one for 7,000 rpm and one for both. To tune for one engine speed we used two different length pipes inserted in between the main header, $1\frac{5}{8}$", and the taper up to the 2" muffler tube. The long step, 3.1" for the 5,000 rpm resonance can be seen in Figure 44. In order to gain a step, a section of each pipe and a flange was cut to make a sudden expansion from the $1\frac{5}{8}$" tube exiting the port to the 2" tube entering the muffler. Those pipes were then welded to the flange to produce the step seen in Figure 43.

Figure 43: Stepped exhaust pipe

Each piece of the exhaust needed to be clamped and sealed but had to be removable. To do this we used traditional U bolt exhaust clamps. Flanges were cut so that one section of pipe would slip into the other. (Far left of Figure 43) The oversized pipes were then slotted in 4 places so they could be collapsed, thus sealing the transition. The U clamps can be seen again in Figure 44. We chose to purchase the straight-through FMF Power Bomb muffler due to its light weight and low price. All the exhaust components can be seen in Figure 45 below.

Figure 45: All exhaust components

7.3 Compression

The piston chosen was a 13.5:1 compression piston manufactured by JE pistons. (Figure 46: JE High Compression Piston) This piston is made of 2618-T6 high tensile forged aluminum then machined to its final dimensions. Compression is gained by raising the height of the piston dome. This reduces the volume between the top of the combustion chamber and the piston increasing compression. It is very easy to see the increased height of the dome as JE had to machine very distinguished pockets into the piston to ensure that the valves do not run into the top.

Figure 46: JE High Compression Piston

Installing this piston is very simple and it comes with all the associated components needed to overhaul the entire system. In the package come the piston, a ring set, a connecting rod pin and the clips to keep that pin located inside the piston. The process of installing the piston begins by removing the old one. To start, the valve cover and camshafts must be removed. (Figure 47) Once the camshafts are removed and the timing chain is hung from the engine the head bolts can be removed. These are typically either studs or in our case torque-to-yield bolts.

Figure 47: Valve cover removed, Camshafts Visible

Torque-to-yield means that there is a torque spec, then they are tightened an additional 180° in order to stretch the bolt. This results in a constant tensional force produced by the bolts applied to the head of the engine keeping it tight and sealed. Because these bolts are stretched so much during installation, they should be replaced after performing this break down.

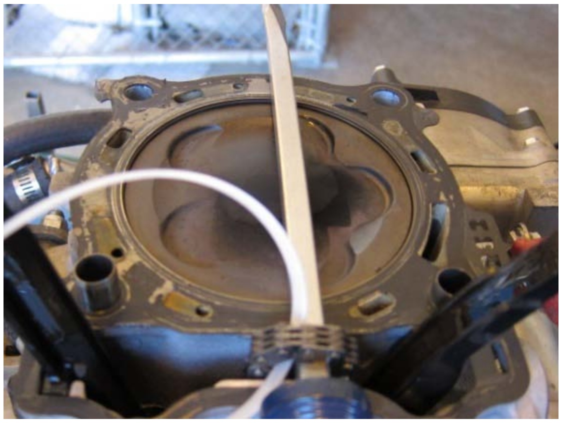


Figure 48: Piston and Cylinder

Once these bolts are removed, the head can be removed revealing the top of the combustion chamber, piston and cylinder.(Figure 48) In our case the valves and piston were a brown color. (Figure 49) This is actually a good color because it means that the engine is not running too rich and leaving un-burnt fuel in the chamber. If the engine had been running excessively rich the inside of the chamber would have been a wet,

Figure 49: Head of engine showing valves and top of combustion chamber

black color. Also, this ensured that we did not have any oil leaks and allowed us to inspect the bottom side of the valves. From the previous fuel map we know that the engine was running around 13.5:1 AFR or a little rich. Once the head is removed the cylinder can be lifted off the rest of the engine leaving the piston with rod exposed. The piston is held in place with a pin that connects the connecting rod to the bottom of the piston. This pin is pushed out after the retaining clips are removed.

Figure 50: Stock Piston. Scored with all ring gaps aligned

Once the piston was off we found that there was some scoring on the outside. We determined this to likely be due to the ring gaps all being aligned with each other. (Figure 50) This allows gases to make their way past the piston heating this area up and causing that area to wear more than all the others.

Once the stock piston was removed we installed the new one. This process began by installing the piston rings onto the piston. These rings were file to fit so they can be custom matched to any application. What file to fit means is that the ring longer than it needs to be and must be filed to a shorter length so that the proper gap can be obtained. To determine the proper gap JE provides a chart. We decided to use a circle track tolerance which should give us a little extra tolerance for thermal expansion of the rings. Once the correct gap was obtained the rings were all installed on the piston.

Figure 51: New Piston on scale

We then weighed the two pistons and found that the stock piston and ring combination was 277 grams while the new piston and rings are 315 grams. (Figure 51) This is likely not a major issue but in an ideal world the engine should be fully balanced to minimize vibration due to the change in weight. The piston was then installed using the new crank pin and clips. (Figure 52) Great care must be taken to ensure that none of the rings are pinched when placing the cylinder back over the new piston. (Figure 53) Once the piston is back in the cylinder the whole assembly can be put back together taking care to tighten all bolts to the proper torque spec and providing the proper lubrication to each component that moves in another.

Figure 52: New Piston installed on Connecting Rod

Before starting the engine it is now essential to add the correct fuel. By increasing the compression the tendency for knock has been greatly increased and must be compensated for by a higher octane fuel. We've decided to use 91 octane pump gas with 108 octane booster which should raise the octane level to 108. Before loading the engine, proper break in procedures must be followed to seat the new piston rings against both the piston and the cylinder walls. This break in procedure is not well established but involves bringing the engine up to temperature slowly then cooling back to ambient. Once this has been performed several times a leak down test should be performed to determine if the cylinder is holding suitable pressure.

Figure 53: Piston in Cylinder

Now that the top of the piston is much taller, when the cams are installed it is absolutely essential to inspect and determine the gap between the valves and the top of the piston. Measurements can be done in a variety of ways, but the best technique involves using clay inside the combustion chamber. After placing a small amount of clay on top of the piston then moving the piston through a stroke the clay can be removed, measured, and the clearance determined.

7.4 Camshafts

After performing the lift calculations we determined that we would use a set of Hot Cams replacement camshafts. (Figure 54) These cams provide approximately 0.009” of extra lift over the stock cams and have a significant duration increase. These cams are a direct stock replacement and would not need to be clearance checked with the stock piston. However, because we increased the compression ratio we now much check clearance before we run the engine with these cams. Also, these cams, like the stock cams do not have any way to adjust cam timing. Since this is a very effective way to modify the torque and power curves we added the Falicon timing adjustment sprockets. (Figure 55) These sprockets give us the ability to adjust timing of the cam shafts and are manufactured specifically to mate with the hot cams camshafts. Detailed instructions on how to install these cam sprockets can be found in Appendix E. Installing essentially just requires pressing the stock gear off and then pressing the new gear on. The only tricky part to this process is pressing the sprocket off of the exhaust cam. This cam has the auto decompression mechanism attached to the sprocket. In order to press this sprocket off a pressing tool was made which straddled the decompression mechanism and pressed on the shaft at the same time.

Figure 54: New Hot Cams camshafts

Figure 55: Falicon replacement Cam Sprockets

Once the camshafts (Figure 56) are installed in the engine we will measure and set the valve clearance with shims and timing with a dial indicator as previously performed and set the cam to our desired position. Several sets of measurement will need to be taken to ensure that the timing is correct. During this process we will need to use extreme caution as we need to detect any clearance issues before we run the engine. In order to predict clearance issues we will place clay on top of the piston and turn over the engine by hand. Once a couple of revolutions have been performed we will then remove the head again and check for any marks on the clay to determine if there is any possibility of interference. Because the sprockets provide a range of adjustment we will do this for both the fully advanced timing and the fully retarded timing. By doing this all in

Figure 56: Hot Cams camshafts with Falicon adjustable sprockets

one shot we can save a lot of time by only removing the head once. While performing the clearance check we will also establish the cam degree angles. Unfortunately the Falicon sprockets do not come with any angle markings so there is no way to determine how much the timing is changing. To determine the timing we will have to measure the lift profile at different cam positions and determine how much the timing is changed. Once all of these steps are completed we can begin testing in the engine.

7.5 Gearing

For the gearing combinations selected the team currently has as 33 and 40 tooth rear sprocket that can easily be mounted on the car. For the 48 tooth sprocket the team will need to purchase an aftermarket sprocket from a company like Sprocket Specialists. Likely this sprocket will be one solid piece. (Figure 57) They will then need to cut the sprocket in half to make it easier to swap when on the car. Depending on the sprocket mounting they may have to make a mount or produce an alternate sprocket mount for this new sprocket.

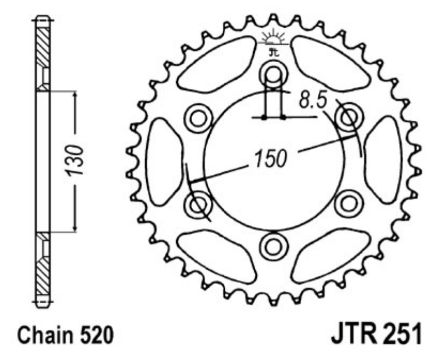


Figure 57: Potential Sprocket

The lead time for the sprocket is likely 1-2 weeks leaving us plenty of time to design and manufacture an adapter if need be. The team currently uses a 520 chain type. This is a thicker chain than some other motorcycles use and it is critical that the sprocket is manufactured at the proper thickness for that chain.

8. Testing

As with design the testing portion of the SPEED System's FormulaSAE development project is broken down into multiple sections, WAVE Model, Dyno and on car testing. Because most of the designs are very theoretical and there are not many formulas or rules of thumb to go by so testing is the main mechanism used to determine trends in component design.

8.1 WAVE Model Verification

In order to use the WAVE model as a legitimate design tool we must verify that it is accurately predicting the output of the engine. To verify this we compiled a list of easily testable changes that can be made while the engine is on the dyno. The current dyno used by the FormulaSAE team is a water brake dyno with a 50 lb load cell. The engine spins the dyno using a chain and two sprockets, exactly the way it powers the car. This makes the power recorded from the dyno very similar to the power actually produced at the wheels by the car.

8.1.1 WAVE Model Verification Testing Plan

While running this test we will be monitoring all the engine systems. We will also be recording the following:

1) Engine speed - Recorded by Dynomite software
2) Torque - Calculated based on load cell value and lever arm
3) Horsepower - Calculated based on Torque and Engine speed

Due to time constraints we decided to use component testing to validate our designs and our WAVE model. The major tests were as follows:

1) Increase intake runner length
    a) When increasing the intake runner length we are investigating how the power and torque curves are affected due to the new "tuned" resonance point. When doing

this test we are looking to see where the curves increase and decrease and by how much. This will quantify how much performance can be gained at various points from just tuning the intake.

2) Remove Plenum
    a) By removing the plenum we hope to quantify how its importance to the car's performance. While running this test we are extremely interested in determining when the power starts to drop off due to choked flow through the restrictor. We expect to see a significant decrease in power from 3500rpm and a sudden drop in power around 8500rpm. These are the points where the flow theoretically will become incompressible and choked respectively without a plenum.
3) Increase Muffler length
    a) By increasing the length of the muffler we will determine how much of an effect the length of the exhaust has on the power curve. We are looking for how the power and torque curves shift as well as how much they shift. We expect that lengthening the exhaust will increase the torque in the lower rpm range.
4) Remove Muffler
    a) By removing the muffler we will quantify what the effect of added restriction in the exhaust is. Also, this will change the length of the exhaust pipe and will change the resonance point. This should be noticeable on the powerband in the form of a spike at a higher rpm.

If by performing the same tests in the model and comparing the curves we can make an educated decision on the accuracy of the model. To be considered a "verified model" it must predict the actual engine behavior in the following ways:

1) Engine speed
    a) The WAVE model must show the same trends as the dyno. If the dyno shows an increase in power or torque at a certain rpm then the dyno must also show an increase in the same area within 5% of the given RPM. Ex. if the dyno shows a horsepower increase from 4000-5000 rpm then the WAVE model must also show an increase beginning from 3800-4200 rpm and ending at 4750-5250 rpm.
2) Horsepower and Torque

a) If the dyno shows an increase or decrease in torque or power, the WAVE model must also show a change in power within 10% of the dyno.

Modeling the restrictor is a known weakness of Ricardo WAVE. If our model can meet these two requirements then it will provide enough accurate information to make engineering decisions from it.

8.2 FSAE Dyno Testing

Dyno testing will be the main testing mechanism used in this project. By using the dyno we can easily compare various component combinations while not having the potentials for inconsistencies in the form of track conditions or driver abilities. Each individual run is plotted in Appendix F. The following provides trends for all compiled runs together.

8.2.1 FSAE Engine Baseline

Before testing any of our components, a baseline has to be established so all future components can be compared to it. This test was completed at constant throttle and was started at the lowest RPM achievable at full dyno load. Once the minimum RPM was established the load was slowly bled off, allowing the engine speed to increase until redline. During this first test, engine speed, brake power and brake torque were recorded. The plot of this run can be seen in Figure 58.

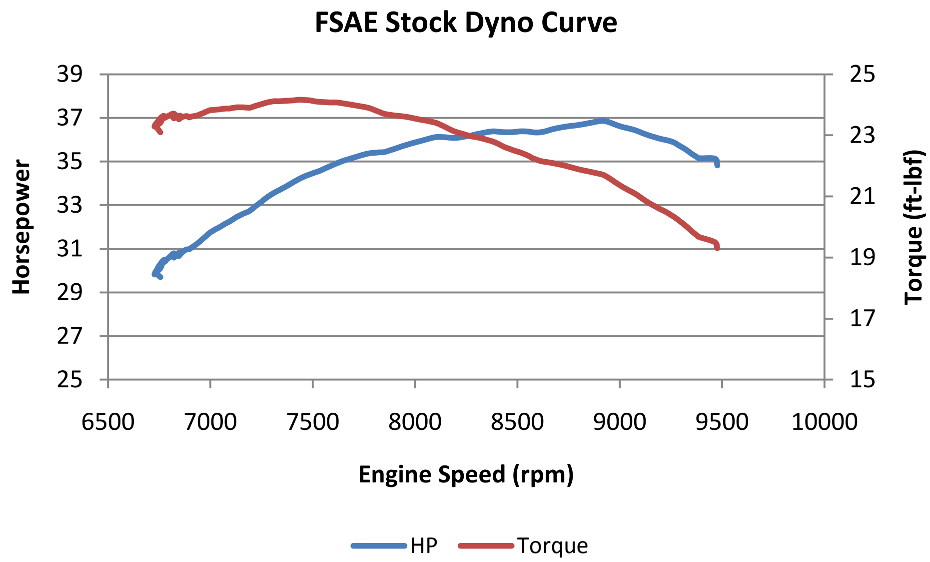


Figure 58: Baseline Engine Torque and Power Curves

In this test power peaked around 37 hp at approximately 8900 RPM. Torque peaked at approximately 24.5 ft-lb$_f$ at 7400 RPM.

8.2.2 Testing Procedure

The testing procedure was established to try and create the most consistent run data we could. This procedure was followed for each test run and is performed as follows:

1. Install desired test components on car and check that all hoses and sensors are installed and tight
2. Turn the engine on and allow to warm up to operating temperature or about 170^{o}F.
3. Once warm, check idle RPM. If above approximately 2800 RPM check for vacuum leaks in intake by spraying starting fluid around all joints in intake.
    a. If engine stalls or bogs, tighten all hoses again to ensure no leaks.
    b. If there is no change in engine speed assume there is no leak and fuel requirement has changed for engine.
4. Place engine into gear and make sure dyno software gear ratio matches the gear being used in the engine.
5. Slowly release the clutch increasing throttle until engine motors dyno with no load.
    a. Check engine speed in MoTec and Dynomax displays
6. Press the “Hold” button on the Dynomax display to activate autoload valve.

a. Autoload valve will increase or decrease dyno load to keep engine at specified RPM. (Typically started at 7000 RPM)

7. Increase throttle to desired level.
8. Use Dynomax to decrease RPM on dyno until the autoload valve is fully open.
    a. This typically happened between 6500 and 6800 RPM.
9. Once RPM has stabilized press the record button on the Dynomax screen and slowly increase the hold RPM. The dyno will decrease the load allowing the engine to rev.
10. Continue increasing hold RPM until redline then bring the RPM back down to avoid bouncing off the rev limiter.
11. Engage the clutch and bring the engine back down to idle and turn the autoload back to "Knob".
12. Allow the engine to idle until engine temperature is under 185° F
13. Turn off engine and plot data.

8.2.3 Intake Testing Results

Figure 60: Short Bell Mouth in Plenum

The following are the results from all intake tests completed on the engine dyno following the preceding procedure. Each test was performed with all the stock FSAE components with exception of the intake. Testing began by using the short bell mouth and plenum. (Figure 60) The runner length for the bell mouth was then adjusted to different lengths to determine how sensitive resonance is to total runner length. Once we had a spread of data a pull was made with the long bell mouth in the short plenum. (Figure 59) This was essentially running the engine without a plenum because the bell mouth was so long that it was completely shrouded by the plenum end cap as seen in. The plenum size was then increased to the medium length, 7in plenum. Similar tests were run with the short bell mouth, then the long. The results

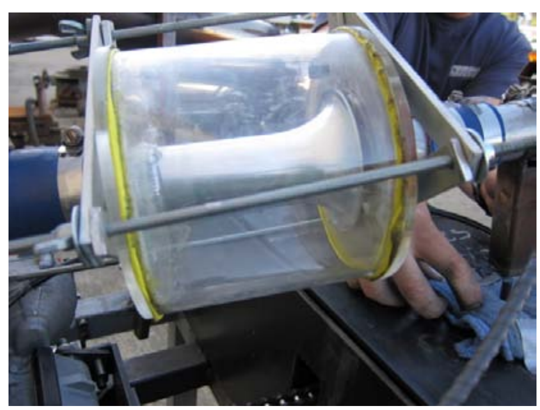


Figure 59: Long Bell Mouth in Plenum

of each test are as follows;

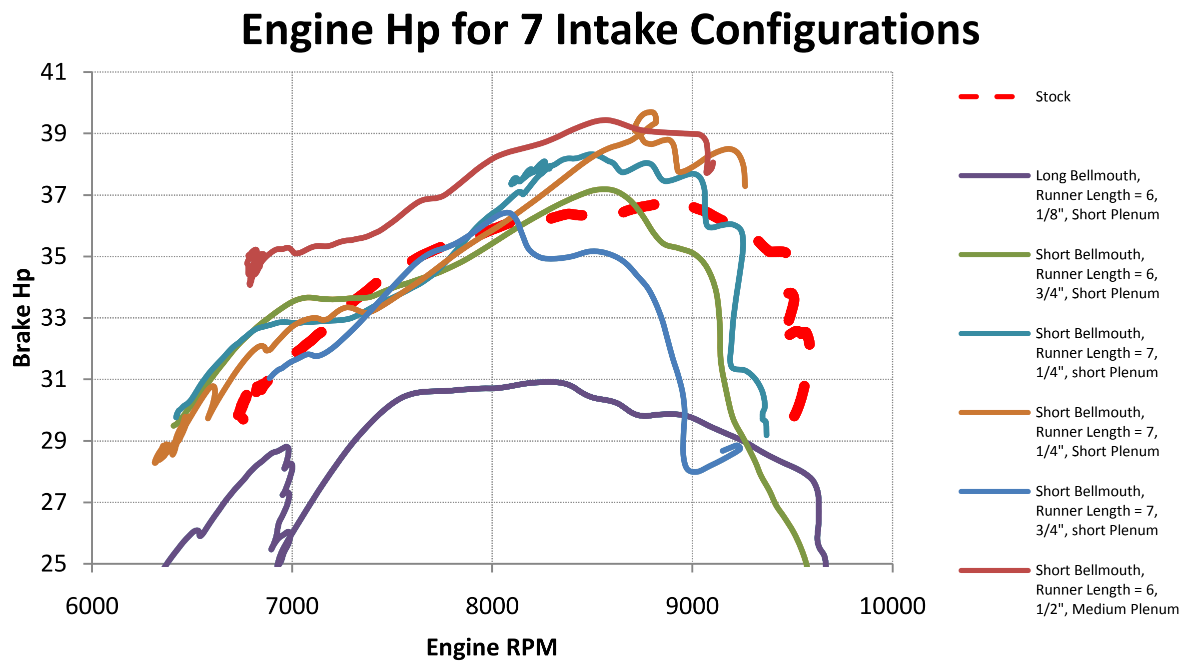


Figure 61: Engine Power for 7 Different Intake Configurations

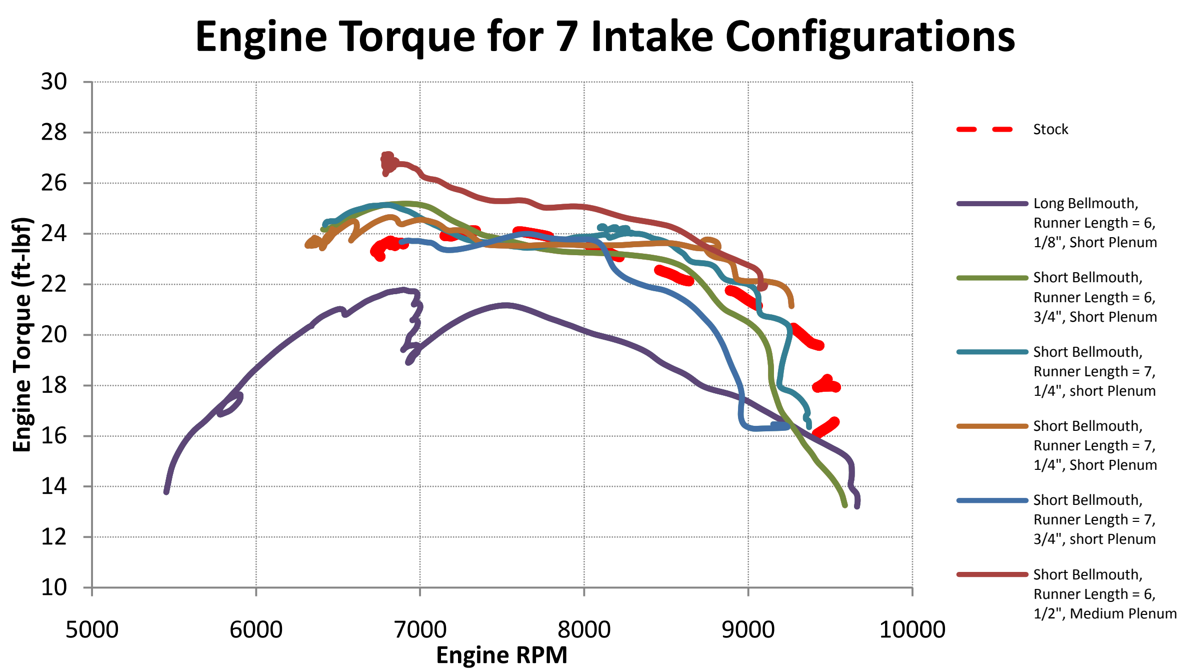


Figure 62: Engine Torque for 7 Different Intake Configurations

From these plots it is easy to determine that the best intake combination is clearly the short bell mouth inside the medium sized plenum. The total runner length was 11.2" and should correlate to a resonance point a little below 9000 RPM. This intake combination produced a maximum of 26.5 ft-lb$_f$ at 6800 RPM and 39.4 hp at 8500 RPM. A full list of individual plots can be found in Appendix F.1. The lower curves in both Figure 61 and Figure 62 show engine performance when there is essentially no plenum. This run was the run performed with the long bell mouth in the small plenum, shrouding the bell mouth. This essentially eliminated the plenum, as the plenum volume wasn't in fluid communication with the rest of the intake. When this happens there is a very significant loss of power with a peak of only 31.9 hp at 8200 RPM and a max torque of only 21.7 ft-lb$_f$ at 6800 RPM. This test confirms that a plenum is absolutely necessary for the success of our application.

8.2.4 Exhaust Testing Results

Once the best intake combination was determined we tested the exhaust. There are three different exhaust configurations, one tuned for 5000 rpm, one tuned for 7000 rpm and one tuned for both using a stepped header. We used the same procedure used in intake testing for exhaust testing. Both the non stepped exhausts were tested twice and the stepped exhaust just once due to time constraints and rain. The results are as follows;

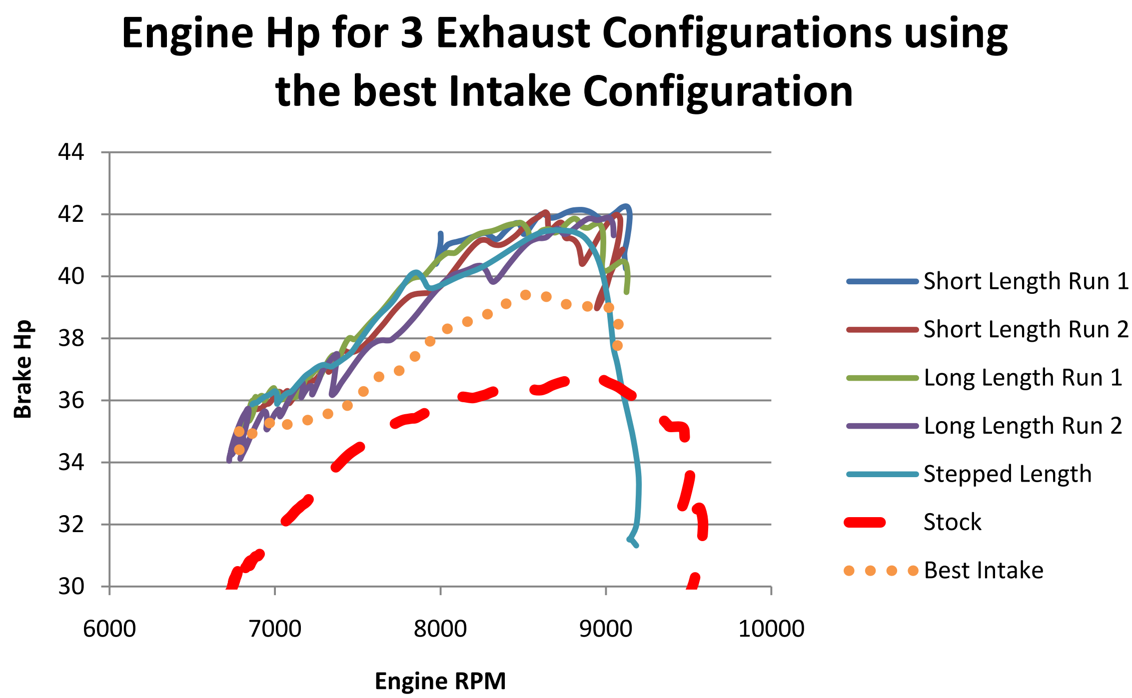


Figure 63: Engine Power for 3 Different Exhaust Configurations, using best Intake Configuration

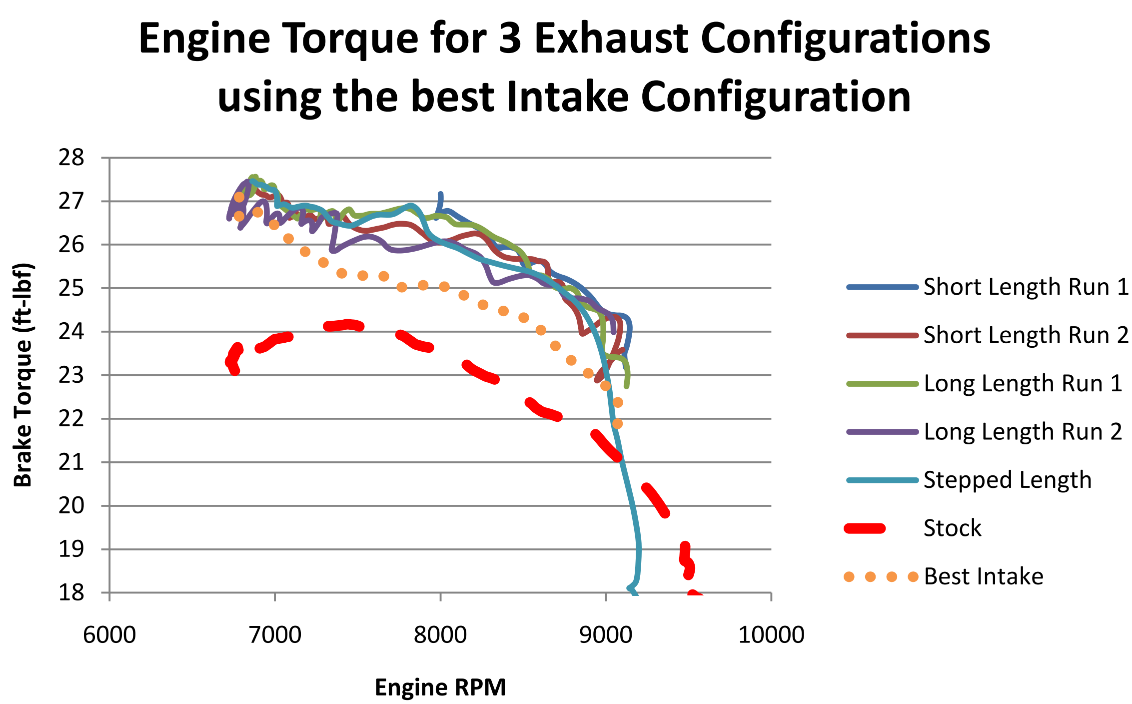


Figure 64: Engine Torque for 3 Different Exhaust Configurations, using Best Intake Configuration

From these runs it is very apparent that a free flowing exhaust makes a huge difference in overall power and torque. Replacing the stock baffle muffler with the straight-through FMF muffler, in addition to the tuned exhaust lengths, dramatically increased the performance of the engine. Although peak torque appears to be very similar to the best intake, the torque across the entire RPM range is increased by up to 2 ft-lb$_f$. Over the stock system, torque is increase by approximately 3 ft-lb$_f$ throughout the RPM range. (Figure 64)

In the power plots it becomes even more apparent how much the free flowing tuned exhaust helps. (Figure 63) With a peak power gain of 3 hp over the best intake and almost 6 hp over the stock setup, the intake and exhaust combination almost met our overall performance goal for the engine by themselves. As with the torque, the power increase appeared across the entire powerband, giving us more power everywhere and not just at one distinct point. As with the bell mouth in the intake, the difference between a good free flowing exhaust and a non- free flowing exhaust was dramatic. However, the differences between each of the free flowing exhausts were very small. A list of all five individual run plots can be found in Appendix F.2.

8.2.5 High Compression Piston Testing Results

Installing and running the high compression piston was the first high risk part of the project. With the higher compression came the higher the chances of incorrect clearances, a significant increase in power and heat but most of all increase the likelihood of knock if not tuned correctly. Knock, as explained earlier is the tendency of the air-fuel mixture to auto ignite, or detonate, before the spark plug ignites the mixture. This creates an extreme pressure pushing the piston back down as it moves up toward the top of the compression stroke. This puts tremendous stress on the mechanical components of the engine and is one of the fastest ways to destroy an engine. To combat the possibility of knock, knock levels were recorded during exhaust testing and compared to the values recorded during piston testing. We observed that knock increased across the RPM range by approximately 10% but never saw a spike indicative of a knocking engine.

The engine started on the first try with the high compression piston, stepped exhaust and medium plenum with the short bell mouth. Once the break-in cycle was complete, the engine was loaded. The current fuel and ignition maps appeared to be very good for the higher compression and did not need adjustment. Three pulls were then measured, each individual run can be found in Appendix F.3 with all three runs plotted in Figure 65 and Figure 66.

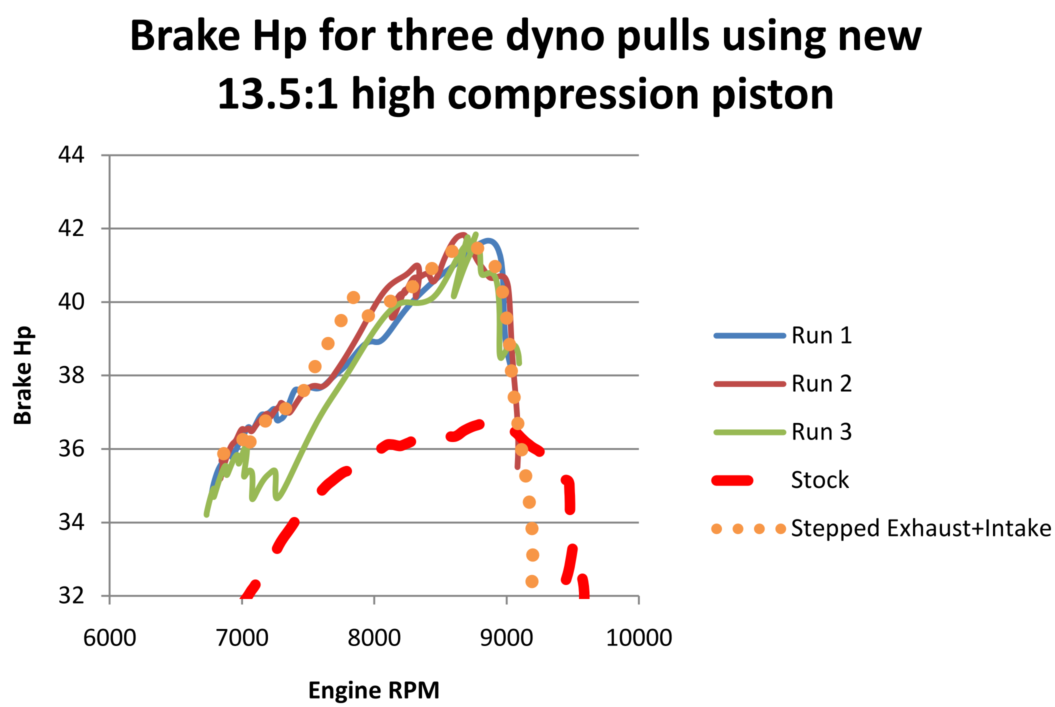


Figure 65: Engine Power using High Compression Piston

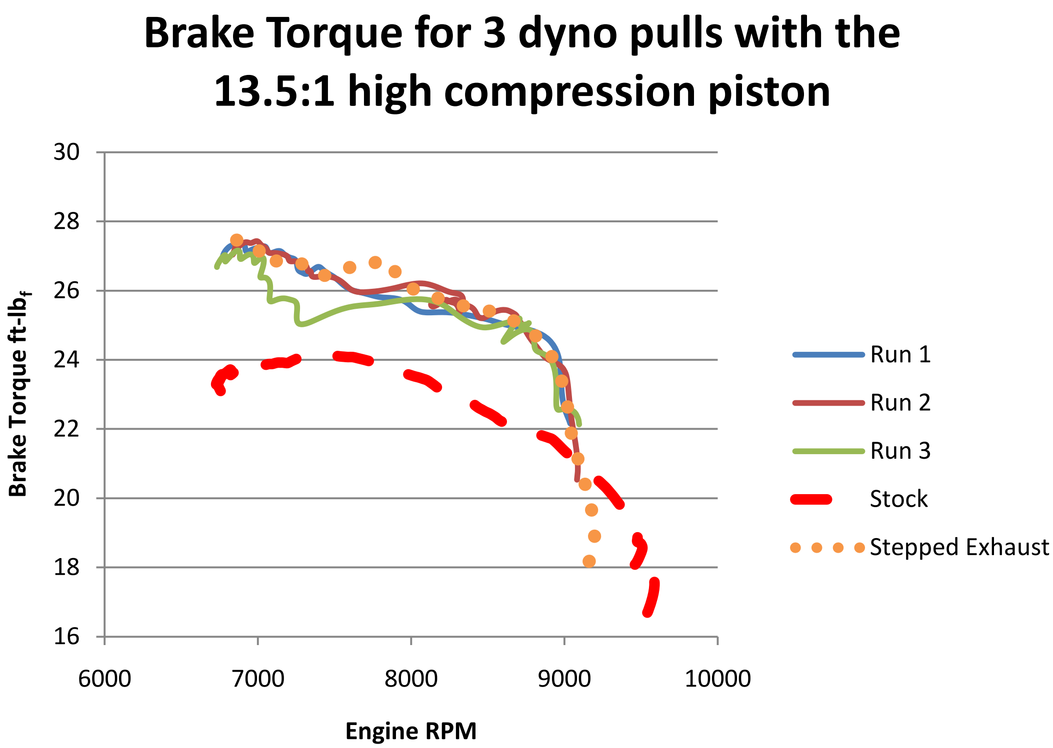


Figure 66: Engine Torque using High Compression Piston

In both these plots, Figure 65 for power and Figure 66 for torque, it is apparent that the 1.2 point increase in compression ratio did not make a significant difference in engine performance. The WAVE model predicted that the increased compression would result in 2.4% and 1.75% increases in torque and power, respectively. This is likely a high estimate as the WAVE model over-predicted the original engine performance by approximately 13%. Another possible explanation for the lack of increase in performance could be that the cam timing is no longer optimal. Because the compression has been increased, the cam timing may need to be changed to achieve the same scavenging effects achieved with the stock piston.

8.2.6 Camshaft Testing Results

Installing and timing the camshafts is a very time intensive exercise. In addition in the FSAE engine there are additional timing sensors that have been placed on and in the vicinity of the cam in order to use it to determine proper engine timing. Before running the cam, it must be degreed in multiple places. First it must be degreed in the stock position. Then it will need to be degreed to plus/minus 12^o in increments of 4^o. At each of these points the cam shafts must both be marked so they can be easily adjusted during testing. Also, during this exercise it must be ensured that the new cams do not cause the valves to interfere with the new piston. Because the new cams push the valves deeper and the dome of the new piston is taller, it is essential to check that the two will not interfere; should the valves "crash" into the piston, or catastrophic engine damage would result. To ensure that the cams do not ever interfere we will check the maximum retarded and advanced cam positions by placing clay on top of the piston and measure how far the clay is depressed by the valves.

At this point in the project, SPEED does not have the time or resources to finish cam testing. It is highly recommended for the team to continue the development and install the cams on the engine. Tuning will likely be required to re-establish a solid baseline.

8.3 Recommended FSAE On-Car Testing

On-car testing is the most difficult part of testing. It is difficult in the sense that it will be difficult to draw strong comparisons between the current vehicle and the vehicle with the SPEED components. Changing parts in the engine will likely take half a day which means that the fastest testing turn around would be the next day. The team should perform testing over two days with the first consisting of current vehicle testing and changing components. The second day would consist of testing SPEED components.

8.3.1 Acceleration Testing Plan

Testing acceleration should be fairly repeatable. The team should set out the 75 meter acceleration course in one of the Cal Poly parking lots where there are no obstacles which could affect the times. They should then run 5 acceleration runs back to back and average the times. If the first 1 or 2 times are significantly slower, then they should be thrown out and run a second time. By averaging five runs we hope to minimize the effect of driver and surface inconsistency between runs.

Once the current vehicle runs its acceleration runs, mark the cone locations with chalk to ensure that we set the course up in the same location. The next morning at approximately the same time the course should be set up again and the tests repeated using the same procedure as the day before.

In addition to acceleration times, driver feedback is very important. We want to watch and understand when the driver is shifting the car, in order to understand if they feel like they have more torque or less torque with the stock setup, and as a result more or less traction.

8.3.2 Autocross Testing Plan

As with acceleration testing, autocross testing should also be performed over a two day period. With autocross the team should record times over 3 sets of 5 laps, averaging each set to eliminate variation. In addition to times, driver feedback in the autocross is very important. If the car is produces too much torque and is unpredictable then the team will need to either tune back the car or change the gearing. The driver should keep in mind the following questions during testing:

1) Is acceleration faster, slower, or equivalent?
2) Is the power delivery predictable?
3) When does power application break tires loose?
4) Is throttle response better, worse or equivalent to the stock car?

The answers to these questions will likely be predictable based on lap times. Faster acceleration with a smooth power delivery and a controllable throttle will make a faster overall car. But if some are better and some are worse then we want to know and try to improve our product.

8.3.3 Other Testing

One of our design criteria was to keep our components to less than 20lbs. To make sure that we met this goal each component needs to be weighed before it goes on to the car. Comparing the component weights against the current components will determine how much weight each system gained or lost.

Reliability testing will not be performed in the timeline of this project. As much as we would like to drive the car for 20hrs or more it is not likely that the team will have that much available time. However, when they do begin to drive on a consistent basis they should record problems that occur with the engine and note the amount of time the engine has been running before each problem to try and fix any problem areas that may arise.

9. Conclusion and Recommendations

The main goal of the project was to achieve a torque curve that is relatively flat and does not peak and then drop off. Based on the baseline dyno pull we hoped to push both the torque and power curves down into the lower engine speeds, in order to gain some of the low-end torque the WR450 has while trying to avoid the restrictor's flow choking at higher speeds. Based on the testing results achieved with our intake, exhaust and high compression piston, it is apparent that we did succeed in moving the torque curve down into the lower speed range. Although it was not possible to load the engine at speeds below 6,500 rpm with the current dyno setup, it is apparent that the curves are shitted when comparing the final power and torque curves to the baseline. When examining the baseline curve, it is easy to see that the curve is rising beginning at 6,000 rpm and peaks around 7,500 rpm. It then drops steadily until the 10,000 rpm redline. The new torque curve begins at its peak, around 6,800 rpm, and slowly decreases to the 10,000 rpm redline. Assuming that the beginning of the curve is the peak then we successfully moved the curve down at least 700-1,000 rpm. In addition to shifting the curve, the maximum torque output was increased to approximately 27.5 ft-lb_f at 6,800 rpm from 24.25 ft-lb_f at 7,500 rpm.

The overall power goal was 45 hp, an increase of 5 hp over the FormulaSAE reported 40 hp with the original intake and exhaust setup. After baselining the original setup we found that it only produced 37 hp at 8,800 rpm. After the completion of high compression piston testing, our system produced a little over 42 hp at approximately 8,800 rpm. This is a 5 hp increase which was the desired hp gain the team was looking for. This 5 hp increase was a 13.5% increase over the baseline, a very respectable increase regardless of the application.

Due to the time constraints placed on the project there are many things that the team should continue to investigate in order to conclude this project. The first is the completion of camshaft testing. As explained in the testing section camshaft testing is very involved and will likely require tuning of the fuel and ignition map. In addition to camshaft testing, additional calibration should be done for the dyno. Currently the load cell was calibrated by hanging a weight from it. In order to calibrate the dyno properly we recommend finding an electric motor with a known torque curve and compare it to the torque curve produced by our dyno. By comparing the two curves the team can find any discrepancies produced while the dyno is rotating. We also recommend getting a smaller sprocket for the dyno. The dyno was unable to produce enough resistance to load the engine below 6,500 rpm. In order to combat this problem the dyno must have the ability to spin faster and create more resistance. During our testing the engine was in 5th gear, its highest gear. Installing a smaller sprocket will allow the dyno to load lower in 5th and potentially allow the team to also run in 4th if they desire to.

In order to help future teams, we recommend that FormulaSAE spend significant time verifying the WAVE model. Once the WAVE model is adjusted to consistently predict engine performance, future teams can use it to investigate the effects of modifications like increased displacement, or even forced induction. The area requiring the most work is the modeling of the restrictor. From experience we know that the engine hp and torque should drop off dramatically once the restrictor begins to choke the intake flow. In the WAVE model the power and torque continue to increase as engine speed increases. This is not correct and could likely be corrected with continued development.

The final recommendation we have is the completion of final vehicle-ready parts out of the correct material. We recommend that the intake plenum be made of aluminum sheet. By producing the entire intake out of a similar material it can be welded, which improves total system rigidity and reduces the possibility of vacuum leaks from portions not sealing. In addition, making the entire intake out aluminum will make mounting much easier. The current mounting for the intake is poor at best due to the acrylic plenums but aluminum plenums will increase the mounting options dramatically. The current exhaust was made of 1018 mild steel because it was cheap and easy to work with and modify. Unfortunately, the heat capacity of this material is very low leaving it prone to cracking and warping, especially in the heat affected zones next to the welds. We would recommend a final exhaust be made of Stainless Steel. Stainless Steel is widely used in high performance applications because of its ability to absorb the extra heat created from running the engine at high load for long periods of time. Manufacturing this exhaust would be a little more time intensive as Stainless Steel requires special welding techniques and it would cost a lot more. However, the benefits of Stainless Steel are well worth the cost and will result in fewer exhaust problems.

With the additions of these recommendations to our current development the 2010-2011 FormulaSAE team could have a very solid engine. The engine development for the WR450 has been very slow up to this point but this senior project has advanced the team leaps and bounds. We feel that we have set the team up for success and have laid the ground work for a very successful engine program in the future.

References

1. Taylor, C., *Internal Combustion Engine in Theory and Practice,* Volume 1, The MIT Press, 1985 pg 171-175

2. Ferguson, C. and Kirkpatrick, A., *Internal Combustion Engines,* John Wiley and Sons, 2001, pg 165-175

3. Blair, G.P. and Cahoon, M., “Best Bell”, *Race Engine Technology.* Sept. 2006

Appendices

Appendix A: Decision Matrix, QFD and Management Plan

Table 16: Decision Matrix

Component	**Weight**	Good Reliability	Power Increase	Low Weight	Low Cost	Feasibility	**Total**
		5	4	3	1	5	
Forced Induction		2	5	1	1	2	44
Intake		4	3	3	4	5	70
Exhaust		4	2	3	4	5	66
Cam Profile		3	2	5	4	4	62
Head/Piston		3	3	5	5	4	67
Variable Valve Timing		2	3	3	1	1	37
Direct Injection		2	3	3	1	1	37
Cooling		4	1	4	3	3	54
Head		5	2	5	1	2	59
Engine Lightening		3	1	5	5	3	54
Custom Gearing		3	1	4	3	4	54

Rank	**Total Value**	**Name**
1	70	Intake
2	67	Head/Piston
3	66	Exhaust
4	62	Cam Profile

Requirement Weight	1	Not Important
	3	Important but not critical
	5	Very Important
Decision Weights	1	Not Effective
	3	Effective
	5	Very Effective

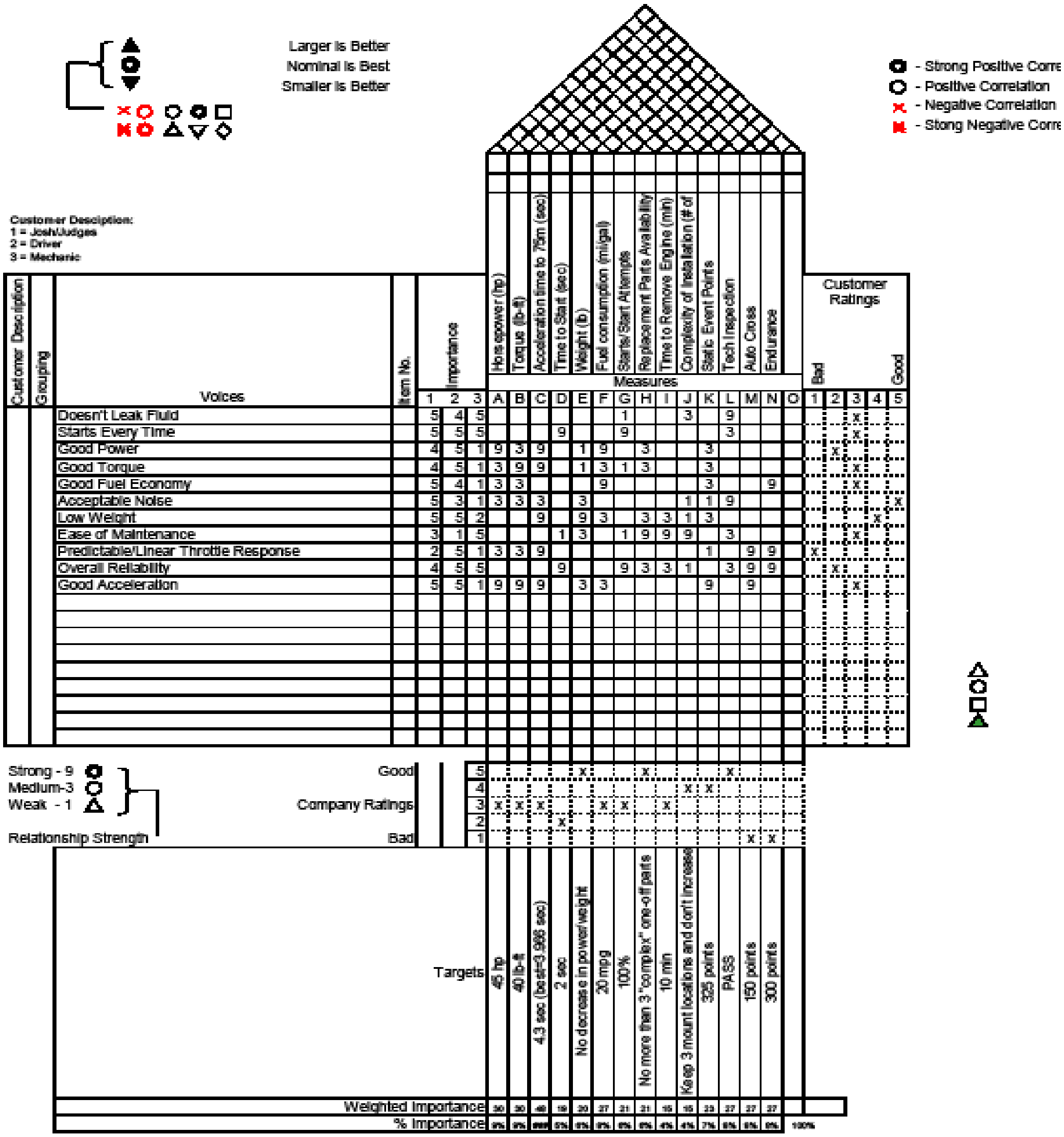
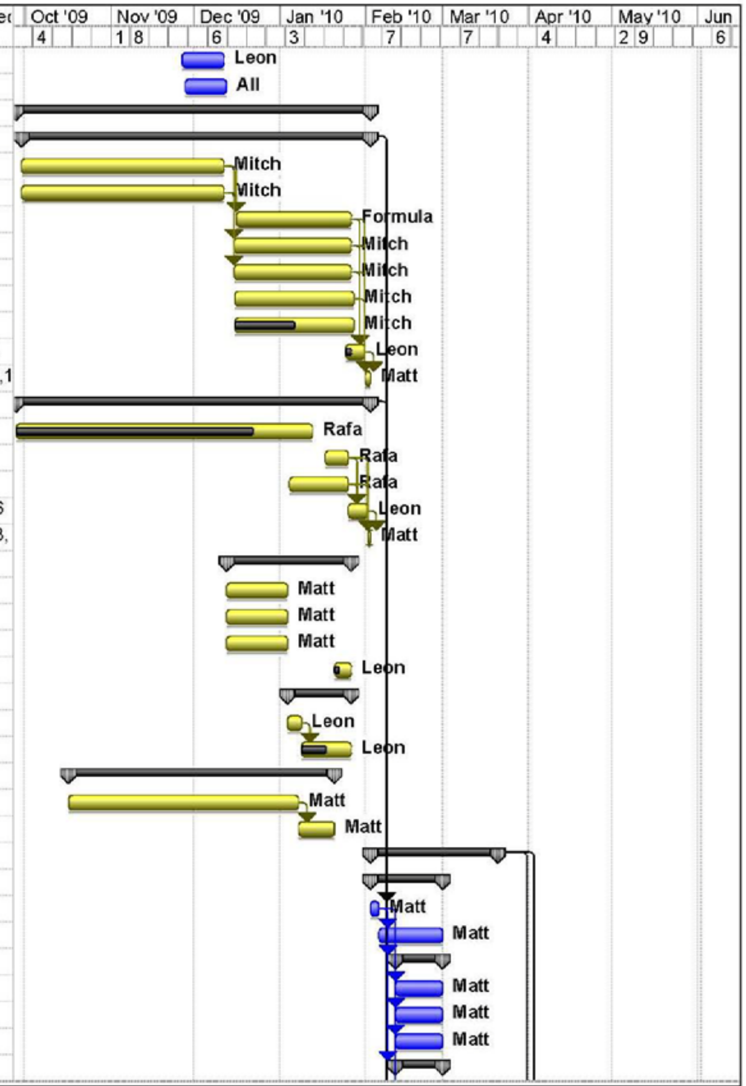


Figure 67: QFD House of Quality

ID		Task Name	Duration	Start	Finish	Predecessors
1		WAVE Model	18 days	Thu 11/26/09	Fri 12/11/09	
2		Research complete	18 days	Fri 11/27/09	Sat 12/12/09	
3		**Design**	**153.8 days**	**Mon 9/28/09**	**Wed 2/3/10**	
4		**Intake**	**151.2 days**	**Wed 9/30/09**	**Wed 2/3/10**	
5		Resonance Calculations	87 days	Wed 9/30/09	Fri 12/11/09	
6		Choked Flow Calculations	87 days	Wed 9/30/09	Fri 12/11/09	
7		Restrictor	50 days	Wed 12/16/09	Wed 1/27/10	5,6
8		Plenum	51 days	Tue 12/15/09	Wed 1/27/10	5,6
9		Runner	51 days	Tue 12/15/09	Wed 1/27/10	5,6
10		Select Throttle Body	51 days	Wed 12/16/09	Wed 1/27/10	
11		Variable Actuation	51 days	Wed 12/16/09	Wed 1/27/10	
12		Test In WAVE	8 days	Mon 1/25/10	Sun 1/31/10	7,8,9
13		Mounting	3 days	Sun 1/31/10	Wed 2/3/10	7,8,9,1
14		**Exhaust**	**153.8 days**	**Mon 9/28/09**	**Wed 2/3/10**	
15		Resonance Calcs	128 days	Mon 9/28/09	Tue 1/12/10	
16		Select Muffler	10.2 days	Sun 1/17/10	Tue 1/26/10	
17		Runner	25.8 days	Mon 1/4/10	Tue 1/26/10	
18		Test in WAVE	8 days	Tue 1/26/10	Mon 2/1/10	17,16
19		Mounting	1.8 days	Mon 2/1/10	Wed 2/3/10	16,18,
20		**Cam Profile**	**53.4 days**	**Sun 12/13/09**	**Tue 1/26/10**	
21		Lift	25.8 days	Sun 12/13/09	Sun 1/3/10	
22		Duration	25.8 days	Sun 12/13/09	Sun 1/3/10	
23		Timing	25.8 days	Sun 12/13/09	Sun 1/3/10	
24		Test in WAVE	7 days	Thu 1/21/10	Tue 1/26/10	
25		**Compression**	**27 days**	**Mon 1/4/10**	**Tue 1/26/10**	
26		Compressoin Study	5.4 days	Mon 1/4/10	Fri 1/8/10	
27		Select Piston	21 days	Sat 1/9/10	Tue 1/26/10	26
28		**Gearing**	**114.6 days**	**Sat 10/17/09**	**Wed 1/20/10**	
29		Analysis	99 days	Sat 10/17/09	Thu 1/7/10	
30		Select Final Gearing	15 days	Fri 1/8/10	Wed 1/20/10	29
31		**Manufacturing Prototypes**	**54.6 days?**	**Wed 2/3/10**	**Sat 3/20/10**	
32		**Intake**	**30.8 days**	**Wed 2/3/10**	**Mon 3/1/10**	
33		Order Materials	3 days	Wed 2/3/10	Fri 2/5/10	4
34		Restrictor	27 days	Sat 2/6/10	Sun 2/28/10	33
35		**Plenum**	**19.8 days**	**Fri 2/12/10**	**Sun 2/28/10**	**33**
36		Cut Plenum Tube to leng	19.8 days	Fri 2/12/10	Sun 2/28/10	33
37		Cut out plenum Endcaps	19.8 days	Fri 2/12/10	Sun 2/28/10	33
38		Bond Ends	19.8 days	Fri 2/12/10	Sun 2/28/10	33
39		**Runner**	**20 days**	**Fri 2/12/10**	**Sun 2/28/10**	**33**

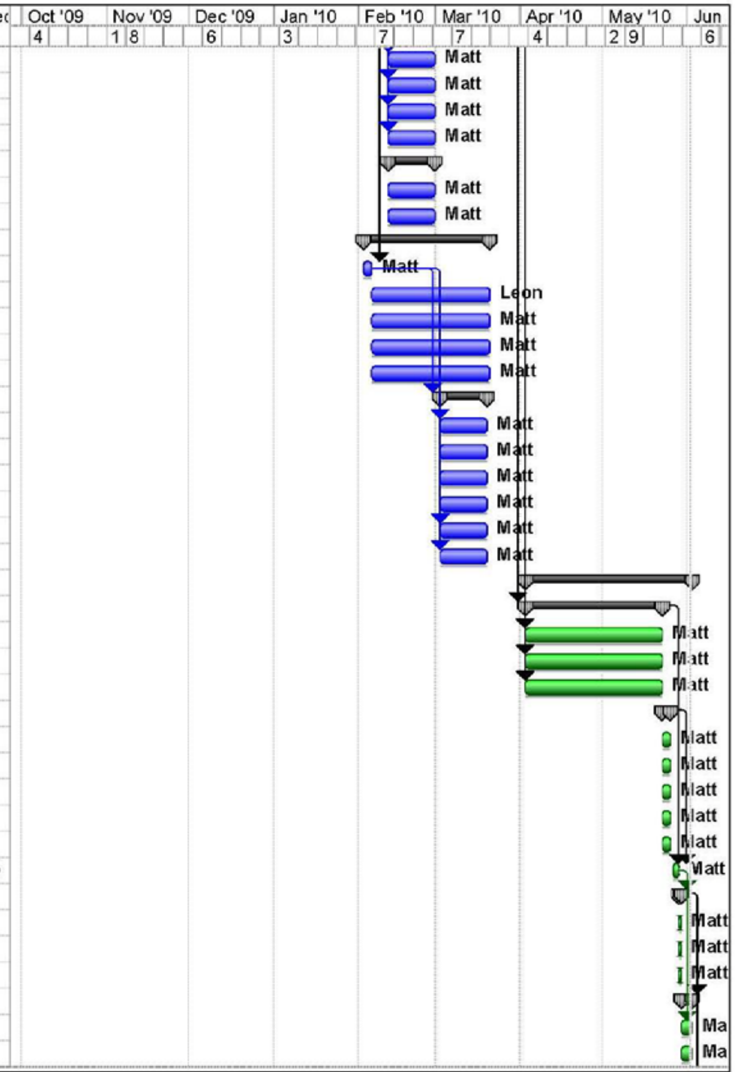


Project: SPEED Gantt Chart
Date: Wed 6/2/10

Task | Split | Progress | Milestone | Summary | Project Summary | External Tasks | External Milestone | Deadline

Page 1

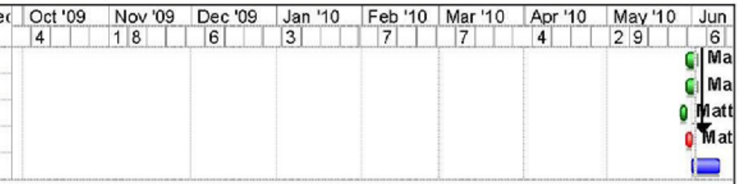
ID	Task Name	Duration	Start	Finish	Predec
40	Cut Tubing	20 days	Fri 2/12/10	Sun 2/28/10	33
41	Weld V	20 days	Fri 2/12/10	Sun 2/28/10	33
42	Throttle Body	19.8 days	Fri 2/12/10	Sun 2/28/10	33
43	Mounting for dyno	19.8 days	Fri 2/12/10	Sun 2/28/10	33
44	**Bell Mouths**	**20.2 days**	**Fri 2/12/10**	**Mon 3/1/10**	
45	Machine	20 days	Fri 2/12/10	Mon 3/1/10	
46	Bond to Plenum	20 days	Fri 2/12/10	Sun 2/28/10	
47	**Exhaust**	**54.6 days?**	**Wed 2/3/10**	**Sat 3/20/10**	
48	Order Materials	3 days	Wed 2/3/10	Fri 2/5/10	14
49	Cut Tubes	51 days?	Sat 2/6/10	Sat 3/20/10	
50	Weld Exhaust Sections	51 days?	Sat 2/6/10	Sat 3/20/10	
51	Assemble	51 days?	Sat 2/6/10	Sat 3/20/10	
52	Muffler	51 days	Sat 2/6/10	Sat 3/20/10	
53	**Runner**	**20 days**	**Wed 3/3/10**	**Fri 3/19/10**	**48**
54	Cut tube 4ft Non Step	20 days	Wed 3/3/10	Fri 3/19/10	48
55	Cut tube 2.5 Non Step	20 days	Wed 3/3/10	Fri 3/19/10	
56	Cut tube 4ft Step	20 days	Wed 3/3/10	Fri 3/19/10	
57	Cut tube 2.5ft Step	20 days	Wed 3/3/10	Fri 3/19/10	
58	Weld Tubeing	20 days	Wed 3/3/10	Fri 3/19/10	48
59	Mounting	19.8 days	Wed 3/3/10	Fri 3/19/10	48
60	**DynoTesting**	**72.6 days**	**Sat 4/3/10**	**Wed 6/2/10**	
61	**Intake Testing**	**59.4 days**	**Sat 4/3/10**	**Sat 5/22/10**	**31**
62	Plenum 1 Test	59.4 days	Sat 4/3/10	Sat 5/22/10	31
63	Plenum 2 Test	59.4 days	Sat 4/3/10	Sat 5/22/10	31
64	Plenum 3 Test	59.4 days	Sat 4/3/10	Sat 5/22/10	31
65	**Exhaust Testing**	**3 days**	**Sun 5/23/10**	**Tue 5/25/10**	
66	Muffler Test	3 days	Sun 5/23/10	Tue 5/25/10	
67	Regular Header test 4ft	3 days	Sun 5/23/10	Tue 5/25/10	
68	Regualr Header Test 2.5ft	3 days	Sun 5/23/10	Tue 5/25/10	
69	Stepped Header Test 4ft	3 days	Sun 5/23/10	Tue 5/25/10	
70	Stepped Header Test 2.5ft	3 days	Sun 5/23/10	Tue 5/25/10	
71	Tune Engine for Intake and Exhau	1.8 days	Thu 5/27/10	Fri 5/28/10	61,65
72	**Compression Testing**	**0.6 days**	**Sat 5/29/10**	**Sat 5/29/10**	**71**
73	Install Piston	0.6 days	Sat 5/29/10	Sat 5/29/10	
74	Check Valve Clearance	0.6 days	Sat 5/29/10	Sat 5/29/10	
75	Test Compresion	0.6 days	Sat 5/29/10	Sat 5/29/10	
76	**Cam Testing**	**4.2 days**	**Sun 5/30/10**	**Wed 6/2/10**	**72**
77	Install New Cam	4.2 days	Sun 5/30/10	Wed 6/2/10	71
78	Test New Cam	4.2 days	Sun 5/30/10	Wed 6/2/10	



Project: SPEED Gantt Chart
Date: Wed 6/2/10

Task | Split | Progress | Milestone | Summary | Project Summary | External Tasks | External Milestone | Deadline

ID		Task Name	Duration	Start	Finish	Predec
79		Test Cam Timing 1	4.2 days	Sun 5/30/10	Wed 6/2/10	
80		Test Cam Timing 2	4.2 days	Sun 5/30/10	Wed 6/2/10	
81		Install All SPEED Components	1.8 days	Fri 5/28/10	Sat 5/29/10	
82		Re Tune Engine For Car	1.8 days	Sun 5/30/10	Mon 5/31/10	72
83		Write Project Final Report	12 days	Tue 6/1/10	Fri 6/11/10	



Project: SPEED Gantt Chart
Date: Wed 6/2/10

Task
Split
Progress
Milestone
Summary
Project Summary
External Tasks
External Milestone
Deadline

Page 3

SPEED Sys BOM

Product	Material	Retailer	Part #	Price	Quantity	Total Price	Notes
Cam							
Cam Shaft Intkae	-	Hotcams	4023-1IN	$ 200.00	1	$ 200.00	
Cam Shaft Exhuast	-	Hotcams	4044-1E	$ 200.00	1	$ 200.00	
Cam Timing Kit	-	Falicon	384-412	$ 213.00	1	$ 213.00	
					Total	$ 613.00	
Gearing							
Sproket 33 Tooth	Aluminum	Formula	-	$ -	1	$ -	
Sproket 40 Tooth	Aluminum	Formula	-	$ -	1	$ -	
Sproket 48 Tooth	Aluminum	Sprocket Specialists	-	$ -	1	$ -	
					Total	$ -	
Compression							
Piston Kit	-	JE/Wiesco		$ 230.00	1	$ 230.00	
					Total	$ 230.00	
Exhaust							
Non Stepped							
1.625" Tubing 6'	1018 Steel	McMaster		$ 75.00	1	$ 75.00	
2" Tubing - U-Bend	1018 Steel	JEGS		$ 20.00	1	$ 20.00	
2 1/4" Tubing - U Ben	1018 Steel	JEGS		$ 20.00	1	$ 20.00	
2 1/4" U Clamp	1018 Steel	JEGS		$ 5.00	2	$ 10.00	6"X6" Plate
2" U Clamp	1018 Steel	JEGS		$ 5.00	2	$ 10.00	
1 5/8" U- Clamp	1018 Steel	JEGS		$ 2.50	2	$ 5.00	
Tapered Tube	1018 Steel	Burns Stainless		$ 25.00	1	$ 25.00	
FMF Muffler	-	E-Bay		$ 170.00	1	$ 170.00	
					Total	$ 335.00	4" OD - 12"
							1'X3'X.25
Intake							
Test Plenum							
Test Plenum (3)	Acrylic	McMaster	8486K398	$ 72.50	1	$ 72.50	
Test Runners	Aluminum	Burn's Stainless	ST-163-17-6	$ 8.25	3	$ 24.75	
Test Bell Mouths (3)	Aluminum	McMaster	8974K971	$ 53.25	2	$ 106.50	
Test Plenum Ends	Acrylic	McMaster	8560K356	$ 31.50	1	$ 31.50	
Throttle Body	-	Formula	-	-	-	-	
Restrictor	-	Formula	-	-	-	-	
Final Intake							
Final Plenum	Aluminum	McMaster	8486K398	$ 13.00	1	$ 13.00	12"X12"X.05"
Final Runner	Aluminum	Burn's Stainless	ST-163-17-6	$ 8.25	1	$ 8.25	
Final BellMouths	Aluminum	McMaster	8974K971	$ 53.25	1	$ 53.25	
Final Plenum Ends	Aluminum	McMaster	8560K356	$ 13.00	1	$ 13.00	12"X12"X.05"
					Total	$ 322.75	
					Grand Total	**$ 1,500.75**	

Appendix B: Tractive Effort Curve Calculations

Vehicle Velocity:

$$\omega_A = \frac{r}{\zeta_P * \zeta_G * \zeta_F} \omega_E \qquad \text{Eq A.1}$$

Where:

ω_A = vehicle ground speed

r = Wheel radius

ζ_P = Primary Gear Reduction Ratio

ζ_G = Gear Ratio

ζ_F = Final Drive Ratio (Sprocket)

ω_E = Engine speed

Tractive force :

$$T_A = \frac{\zeta_P * \zeta_G * \zeta_F}{r} T_E \qquad \text{Eq A.2}$$

Where:

T_A = Force Produced By Wheels

r = Wheel radius

ζ_P = Primary Gear Reduction Ratio

ζ_G = Gear Ratio

ζ_F = Final Drive Ratio (Sprocket)

T_E = Engine Torque

Elapsed Time Between velocity points:

From dynamics, F=ma. Where a, acceleration, can be represented by dV/dt

$$F = m\frac{dV}{dT} \qquad \text{Eq A.3}$$

This equation can then be modified to the form of,

$$dt = \frac{m*dV}{F} \qquad \text{Eq A.4}$$

Or

$$(\Delta t) = \frac{m*(\Delta V)}{F} \qquad \text{Eq A.5}$$

Appendix C: Camshaft Calculations and Measurements

C.1 Valve Train Measurements and Dynamics

Table 17: Stock Cam Profile Measurements

	Crank Angle	Intake Lift (in)		Crank Angle	Intake Lift (in)
	-50	0.001		112	0.345
	-32	0.01		124	0.34
	-24	0.02		134	0.33
	-20	0.03		141	0.32
	-17	0.04		148	0.31
	-14	0.05		153	0.3
	-11	0.06		157	0.29
	-8	0.07		161	0.28
	-5	0.08		166	0.27
	-3	0.09		169	0.26
TDC	0	0.1		173	0.25
	3	0.11		176	0.24
	5	0.12	**BDC - 1**	180	0.23
	8	0.13		184	0.22
	10	0.14		186	0.21
	13	0.15		189	0.2
	15	0.16		192	0.19
	18	0.17		195	0.18
	22	0.18		198	0.17
	25	0.19		200	0.16
	27	0.2		203	0.15
	30	0.21		205	0.14
	34	0.22		208	0.13
	37	0.23		211	0.12
	40	0.24		214	0.11
	43	0.25		216	0.1
	47	0.26		219	0.09
	51	0.27		222	0.08
	55	0.28		225	0.07
	59	0.29		228	0.06
	64	0.3		231	0.05
	69	0.31		234	0.04
	75	0.32		237	0.03
	83	0.33		241	0.02
	94	0.34		246	0.01
				270	0.001

C.2 Valve Lift Calculations

$$Z = \frac{\frac{\pi}{4} b^2}{\bar{A}_f} \frac{\bar{U}_p}{c_i}$$

Where:

Z = Mach index

b = Piston Bore

$\bar{U}_p$ = Mean Piston Speed

$\bar{A}_f$ = Mean Effective Area

c_i = Speed of Sound in Intake air

FIRST CALL $\bar{U}_p$ + C_i

$C_i = (\gamma R T_0)^{1/2} = (1.4 \cdot 287 \cdot 338.7)^{1/2} = 369$ m/s

$\bar{U}_p = 2 \underset{\uparrow \text{ STROKE}}{S} \cdot \frac{RPM_{MAX}}{60 s/min}$

$= 2(\cancel{2.5IN}) \frac{10{,}000 \text{ REV/MIN}}{60 \text{ SEC/MIN}} = 21.13$
(0.0634m)

* 10,000 IS LAST YEARS LIMIT, WANT THIS ~~YEAR~~ TO BE ABLE TO SUPPLY ABOVE + BEYOND

AKA. DON'T WANT VALVES TO LIMIT FLOW

$= 2(0.0634m) \frac{12000 \text{ REV/MIN}}{60 \text{ SEC/MIN}} = 25.36$ m/s

$\bar{A}_i = (1.3)(0.095)^2 \frac{25.36}{369}$

$\bar{A}_i = 0.000806$

OR 8.06×10^{-4} m^2

1/9/2010

MORE VALVES : FIND MAX LIFT

FROM TAYLOR - Pg 175 VOLUME 1

E_V (VOLUMETRIC EFFICIENCY) FALL DRAMATICALLY W/

MACH INDEX (Z) GREATER THAN 0.6

FIG 6-14 PG 175 SHOWS E_V FOR VARIOUS Z

FROM $Z = .6 \rightarrow Z = .3$ $E_V \approx 1$ RAISES W/ $Z < .3$

$$Z = \frac{\frac{\pi}{4} b^2 \bar{u}_p}{\bar{A}_f \, c_i}$$

EFFECTIVE AREA $\rightarrow$ $$\bar{A}_{f_i} = \frac{\frac{\pi}{4} b^2}{Z} \frac{\bar{u}_p}{c_i}$$ (INTAKE)

WHERE $Z \leq 0.6$

SEE Pg 40 FOR CALC W/ $Z = 0.6$

~~2 EQ FOR EFFECTIVE AREA~~ -

$A_f = C_f A_V$

$A_V = \pi d l$

DIAM ↑ ↖ LIFT

FOR EXHAUST :

$$\frac{\bar{A}_e}{\bar{A}_i} = \left(\frac{T_i}{T_e}\right)^{1/2}$$ ASSUMES $Z \leq 0.6$

VALVE RESULTS

Diam Intake (m)	0.027
Cf	0.35
Bore (m)	0.095
Stroke (m)	0.0634
Max RPM	12000
Up (m/s)	25.36
Intake Temp (F)	150
Intake Temp (K)	338.7
ci (m/s)	369
Exhaust Temp (F)	1200
Exhaust Temp (K)	922
Diam Exhaust (m)	0.028

1/9/2010

Intake

Z	Ai	Avi	lift (mm)	lift (in)
0.6	0.000812	0.002320	9.116	0.359
0.5	0.000974	0.002784	10.939	0.431
0.4	0.001218	0.003480	13.674	0.538
0.3	0.001624	0.004639	18.232	0.718
0.2	0.002436	0.006959	27.348	1.077
0.1	0.004871	0.013918	54.696	2.153

Exhaust

Ai	Ae	Ave	Lift (mm)	Lift (in)
0.000812	0.000492	0.001406	7.992	0.315

LIFT OF .36IN FOR 12000 RPM FOR INTAKE

EXHAUST, SOME WHERE AROND .315 OR .32

* LARGER WOULD GIVE BETTER PERFORMANC IN COLD CONDITIONS

ACCORDING TO F+K PG 170 (BOTTOM)

STD PRACTICE IS EXHAUST AREA 80-70% OF INTAKE

$\Rightarrow A_e = .7 A_i$

$A_e = .7(0.000812) = 0.000568$

$\Rightarrow$ LIFT $\approx$ .36in @ 12000 RPM

~~For 80% L=~~

Appendix D: Intake Diagrams

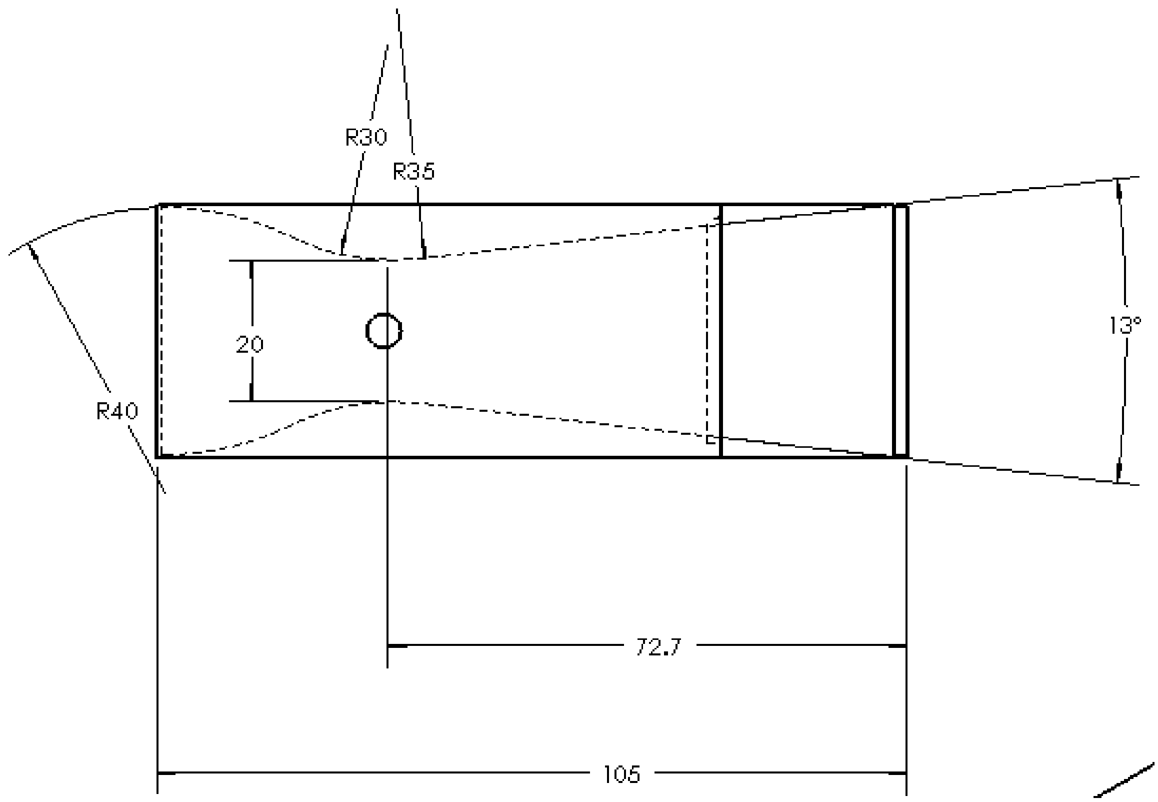


Figure 68: Current design of restrictor. Venturi nozzle shape leaves little room for improvement.

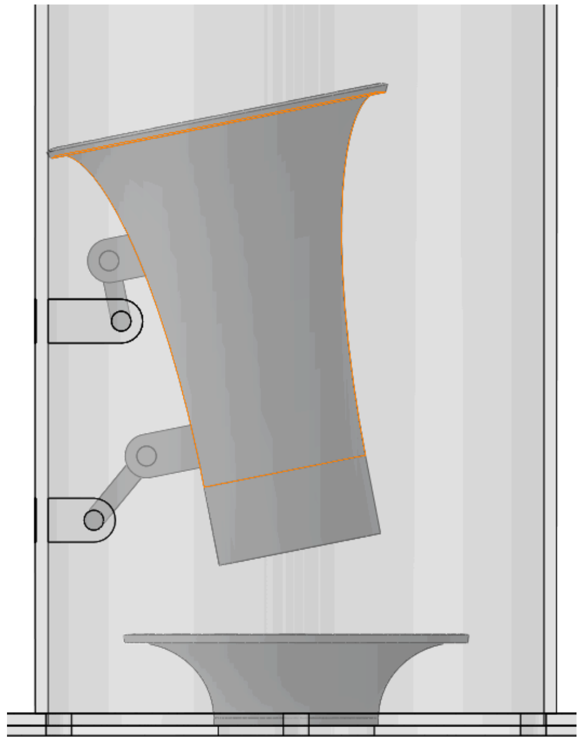


Figure 69: Tilting Bell Mouth Design, High Speed (short) Configuration

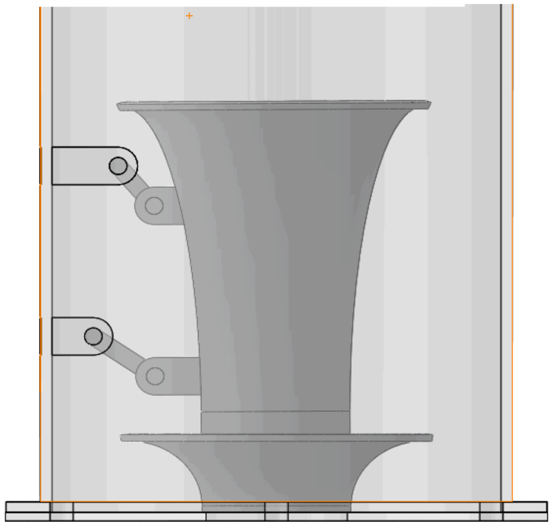


Figure 70: Tilting Bell Mouth Design, Low Speed (long) Configuration

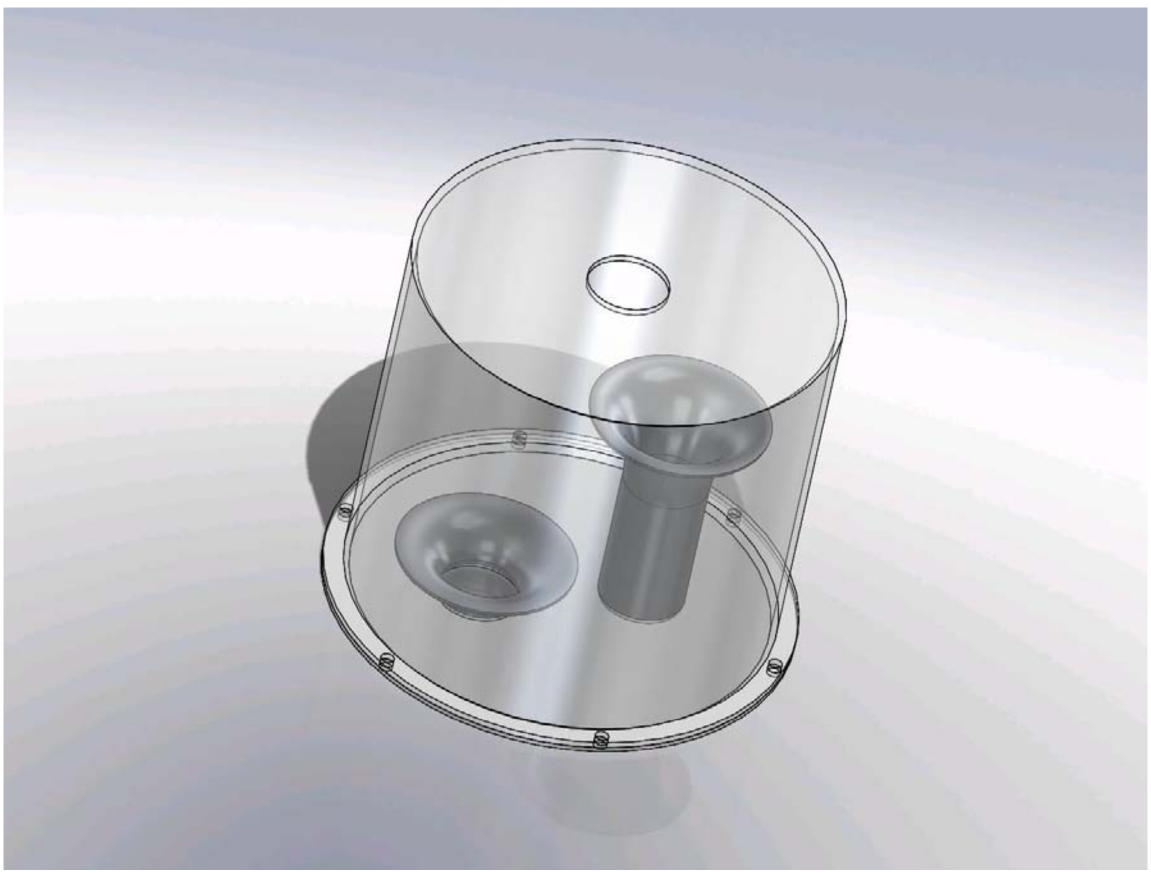


Figure 71: Design for Test Plenum with Selectable Bell Mouths

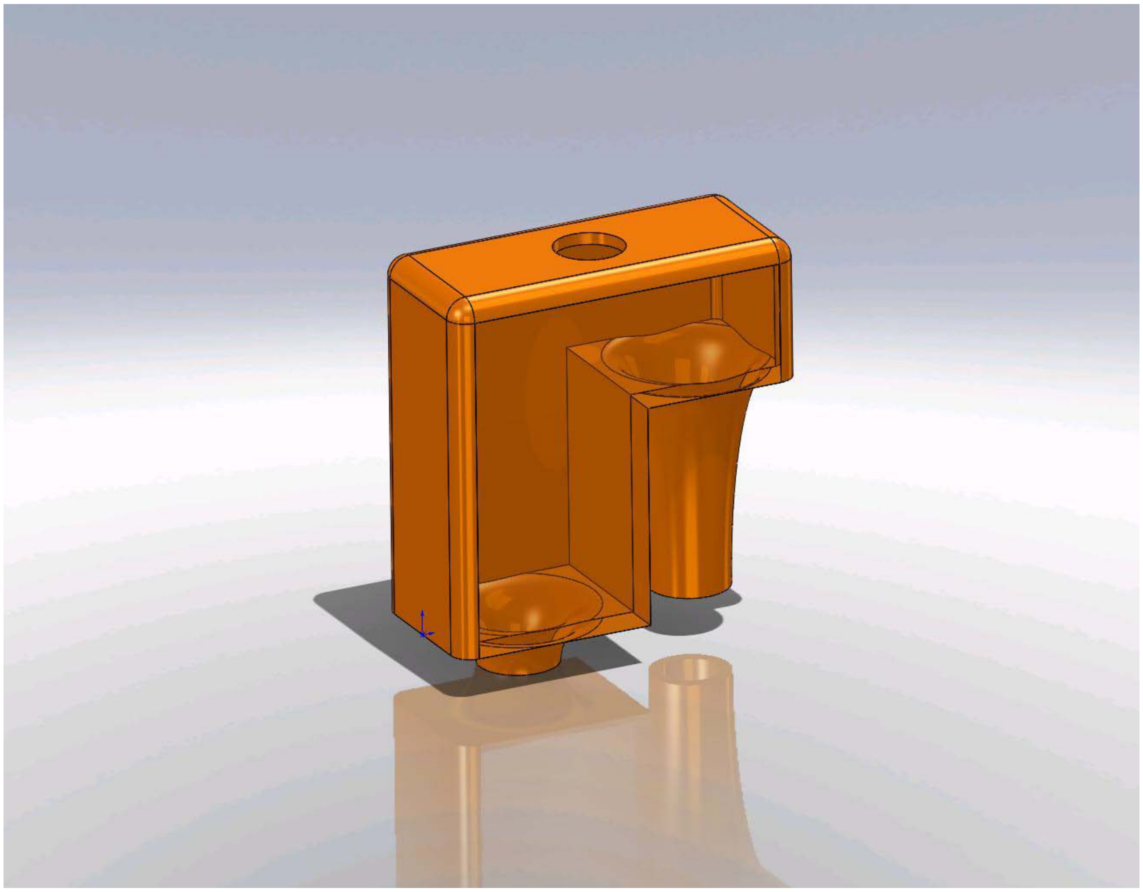


Figure 72: Design for One Piece Plenum with Integrated Selectable Bell Mouths

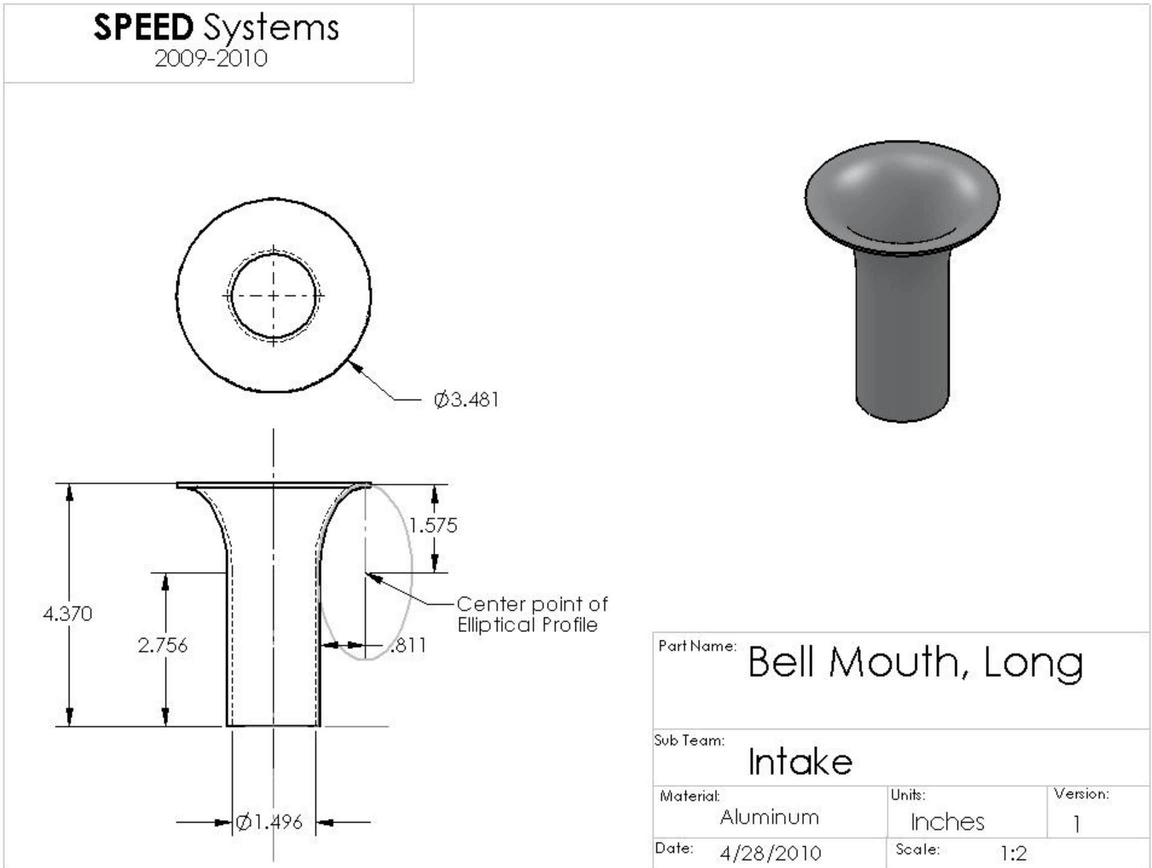


Figure 73: Drawing of Long Bell Mouth

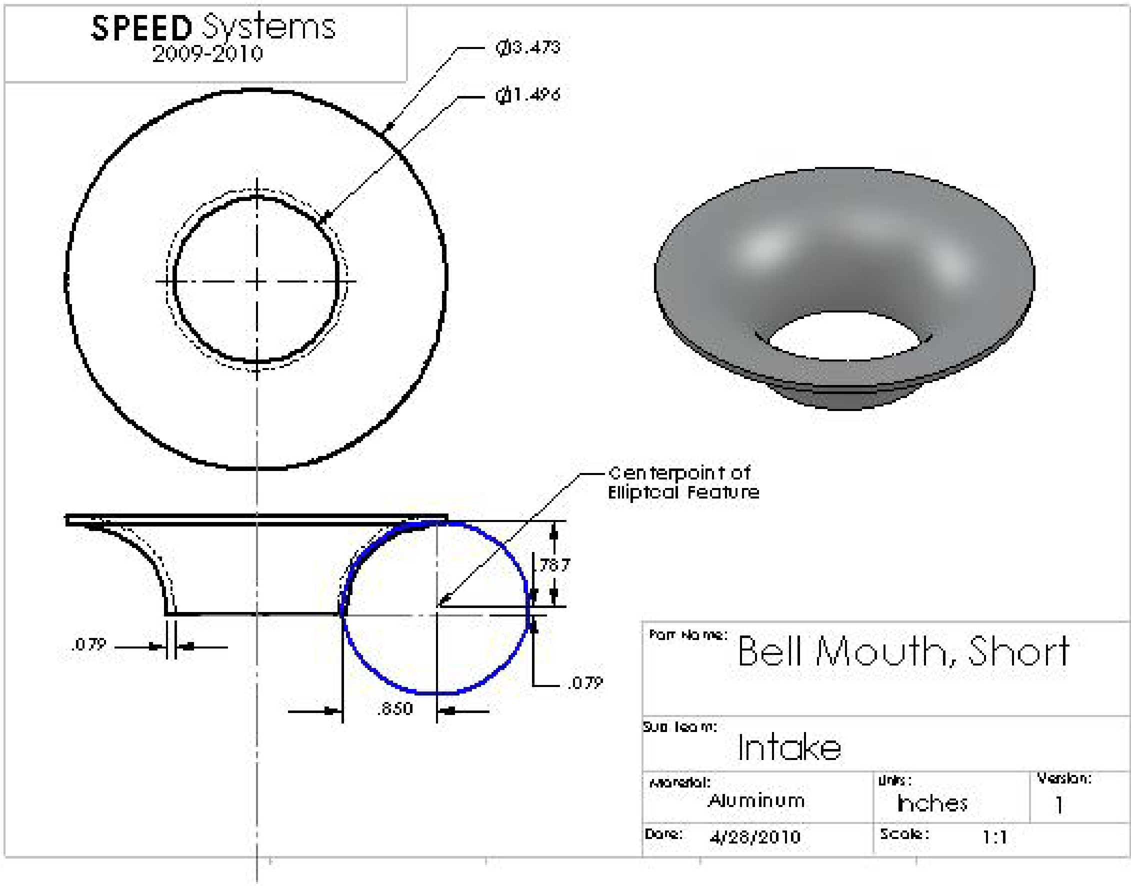


Figure 74: Drawing of Short Bell Mouth

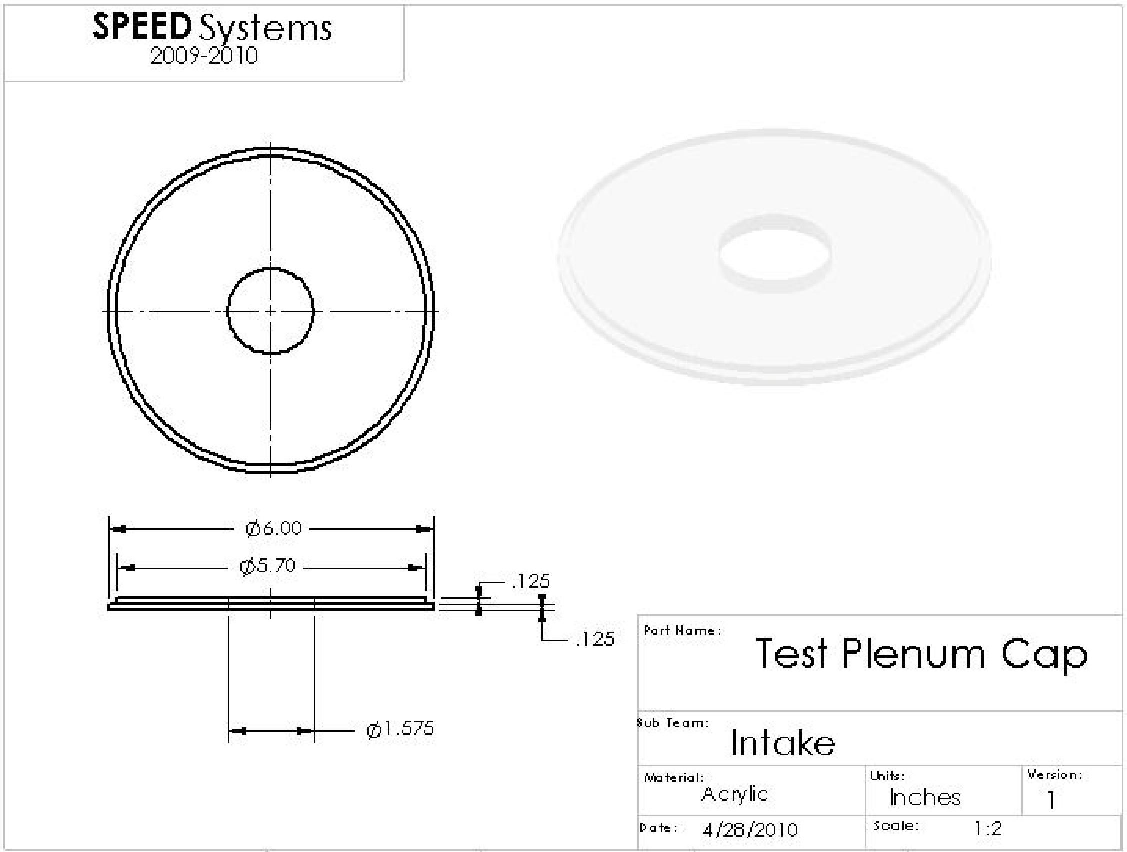


Figure 75: Drawing of Test Plenum Cap (all configurations)

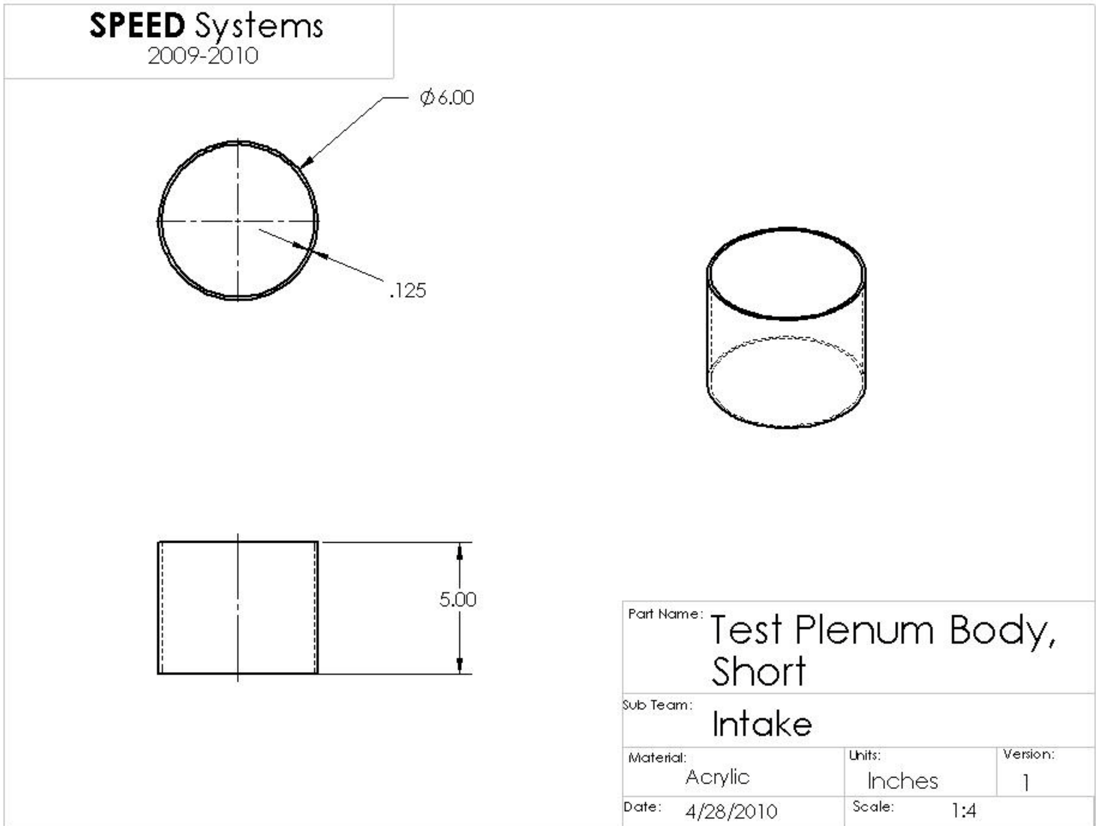


Figure 76: Drawing of Short Test Plenum Body

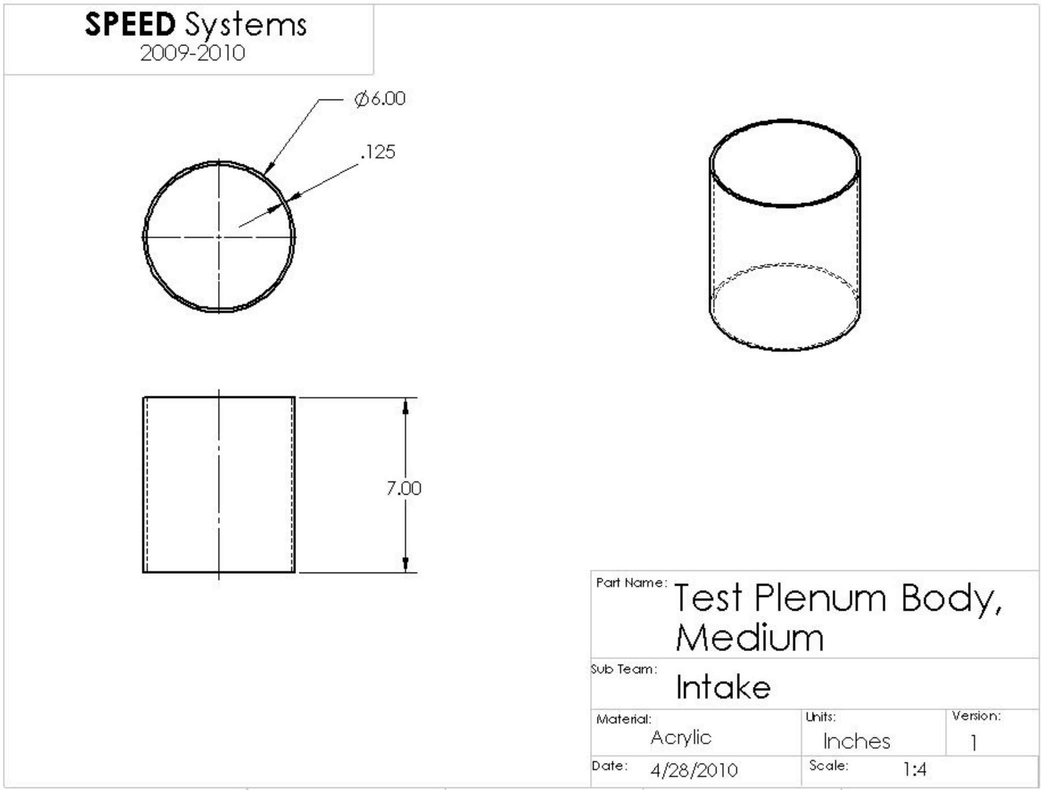


Figure 77: Drawing of Medium Test Plenum Body

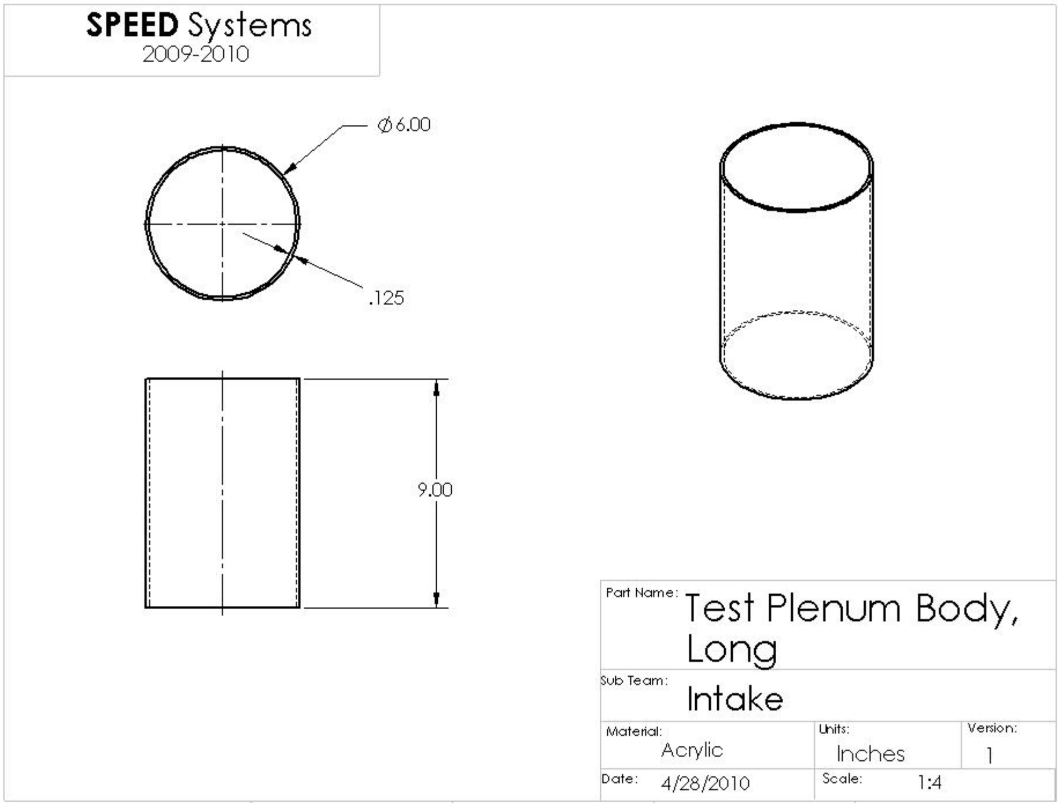
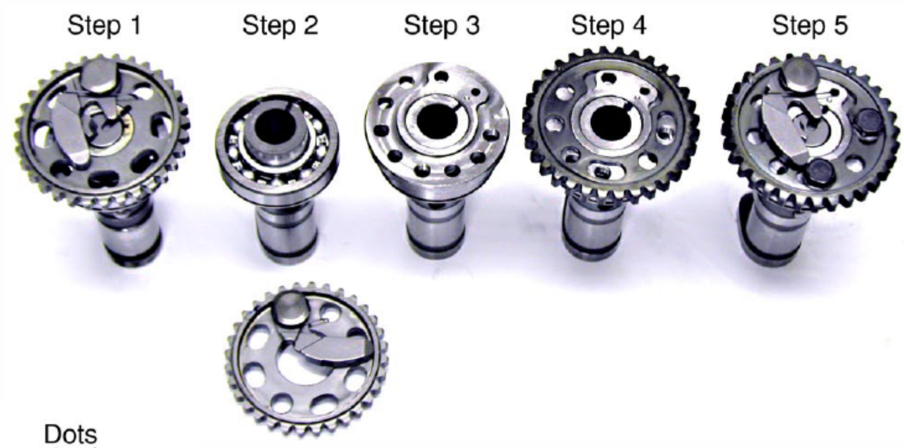


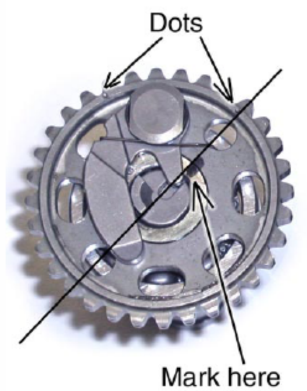
Figure 78: Drawing of Long Test Plenum Body

Appendix E: Falicon Cam Sprocket Installation Instructions

FALICON

Installation Instructions
YFZ/YZ-450 and YZ-250
Exhaust Cam Sprocket





Step 1

Step 1 - Place a straight edge from the Sprocket Alignment Dot, across the sprocket to the opposite tooth. Mark the camshaft with an ink marker.

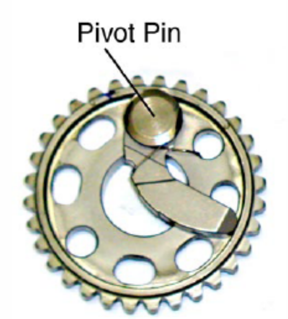
Step 2 - Press off the OEM sprocket. Be careful not to break the arm on the counterweight.

Step 3 - Place the Falicon Sprocket Adapter on the camshaft, aligning the dot to the mark on the camshaft. Press on the adapter.

Step 4 - Place the Falicon sprocket on the adapter, and bolt it in place.

Step 5 - Remove the parts from the OEM sprocket by pressing the Pivot Pin out. Install the Weight, Spring and Pivot on the Falicon Sprocket Adapter by pressing the Pivot Pin in place.

Step 3



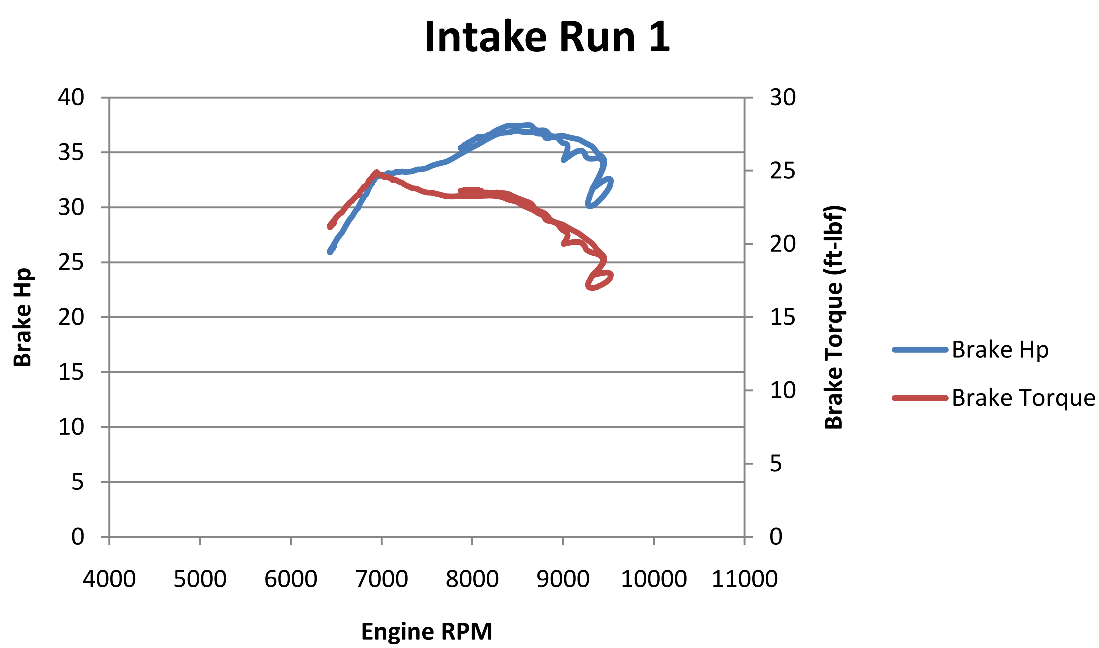
OEM Sprocket


Final Assembly

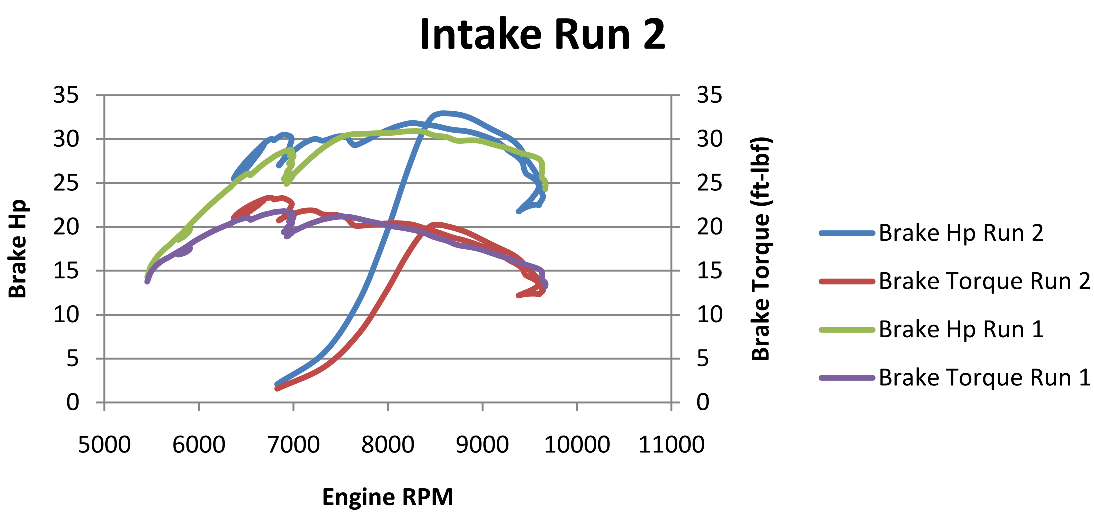
Appendix F: Full Testing Results

F. 1 Individual Intake Test Results:

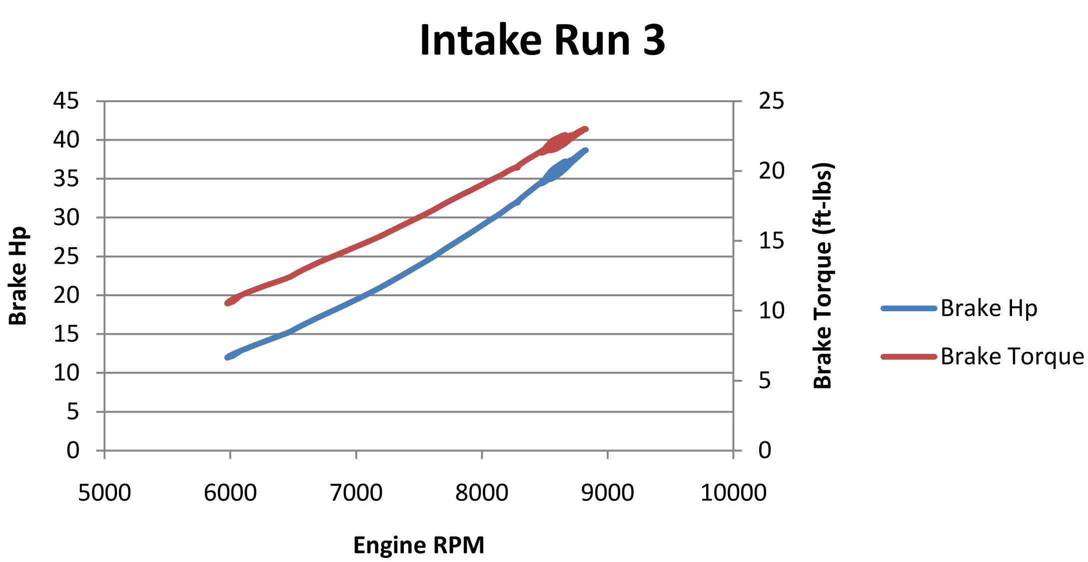
Configuration – Run 1
Port - Short Bellmouth Length = 6, 1/8", Short Pleunm
Short Plenum
4th gear - 20 Tooth Sprocket



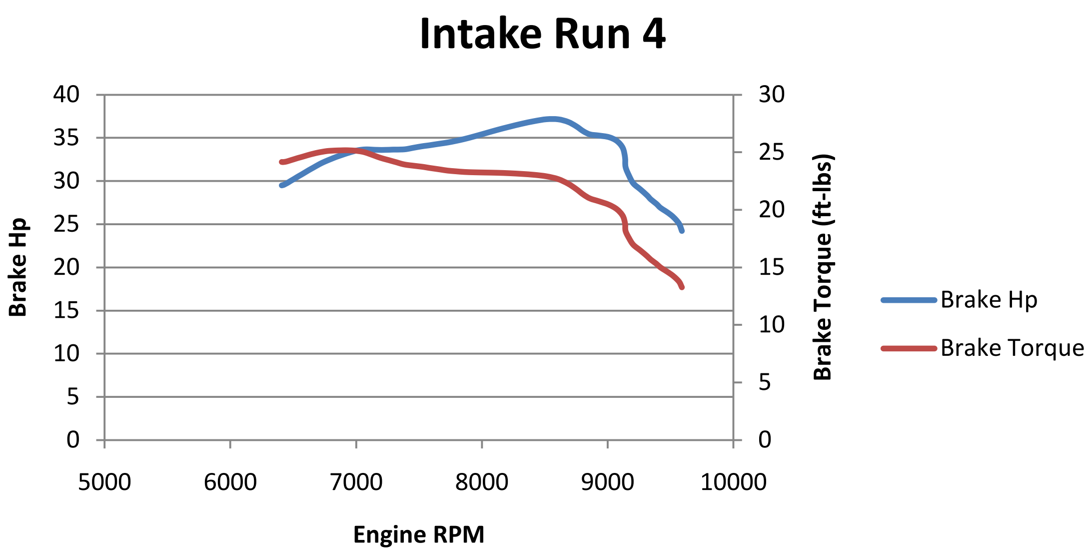
Configuration - Run 2
Port - Long Bellmouth Length = 6, 1/8", Short Plenum
Short Plenum
Long Bell Mouth
4th gear - 20 Tooth Sprocket



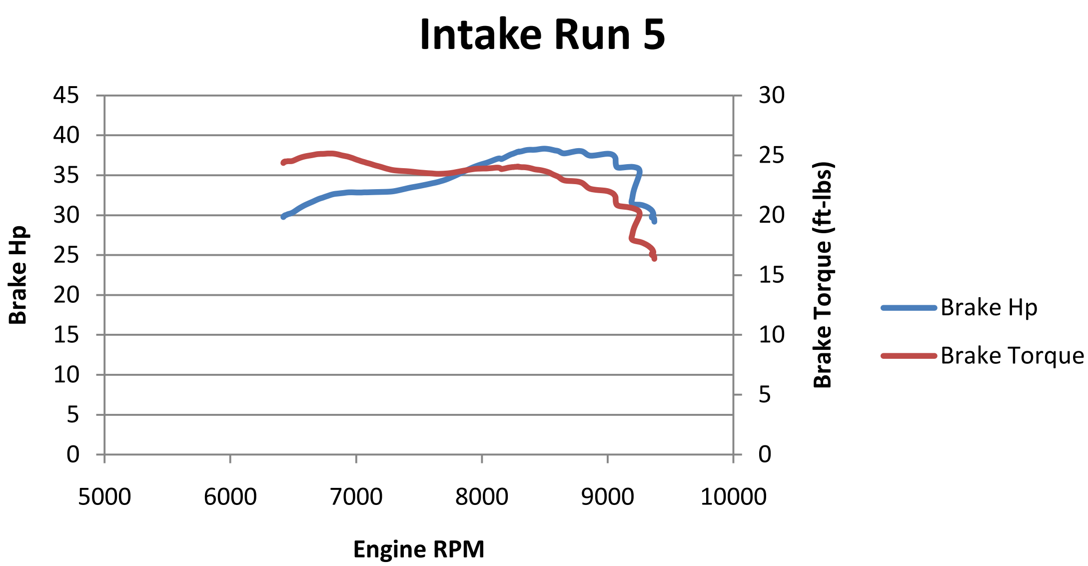
Configuration - Run 3
Port - Short Bellmouth Length = 6, 3/4", Short Plenum
Short Plenum
4th gear - 30 Tooth Sprocket



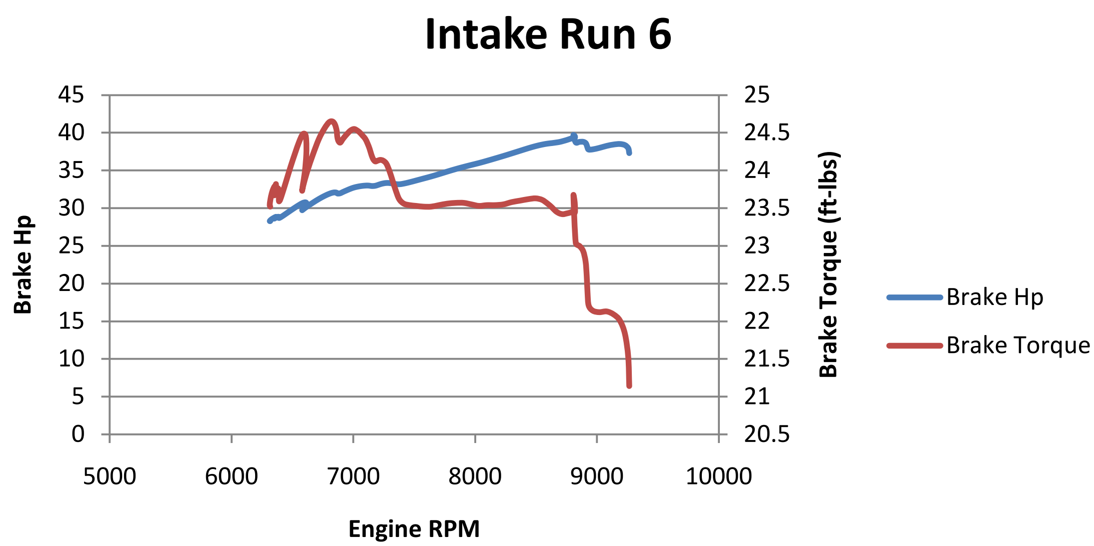
Configuration - Run 4
Port - Short Bellmouth Length = 6, 3/4", Short
Plenum
Short Plenum
5th gear - 30 Tooth Sprocket



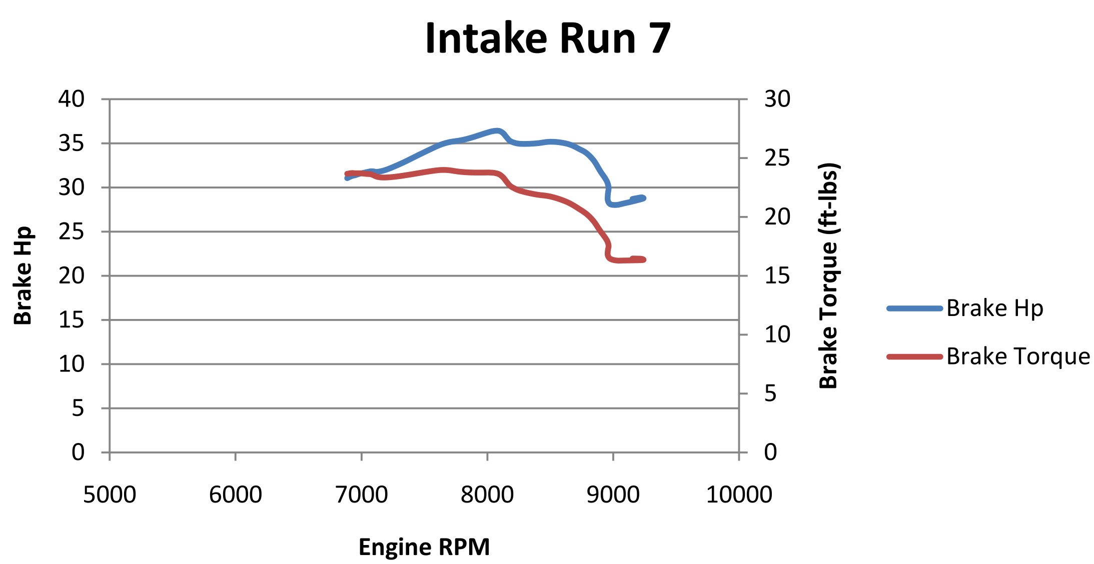
Configuration - Run 5
Port - Short Bellmouth Length = 7, 1/4", short
Plenum
Short Plenum
5th gear - 30 Tooth Sprocket
70% Throttle



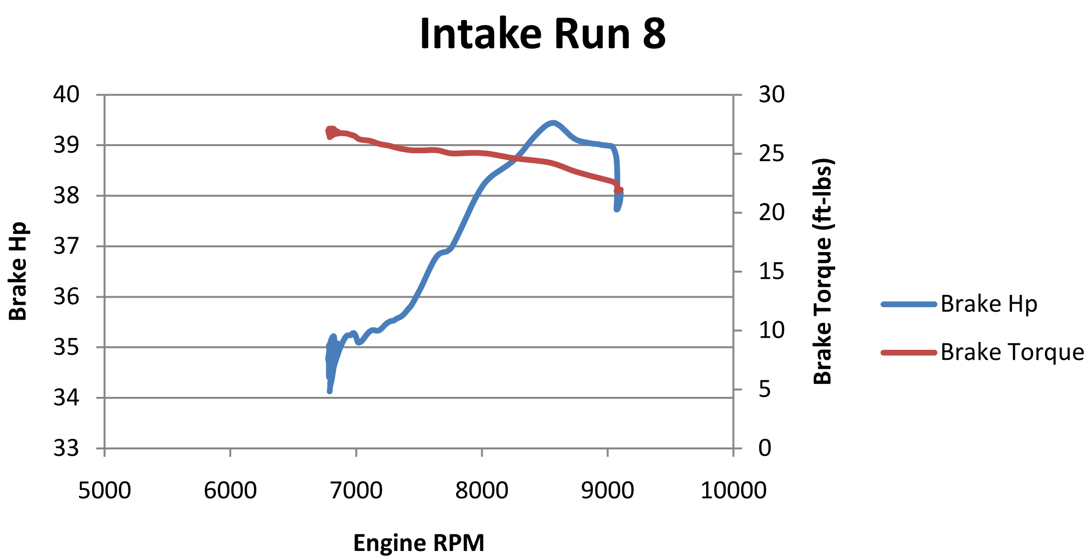
Configuration - Run 6
Port - Short Bellmouth Length = 7, 1/4", Short Plenum
Short Plenum
5th gear - 30 Tooth Sprocket
96% Throttle



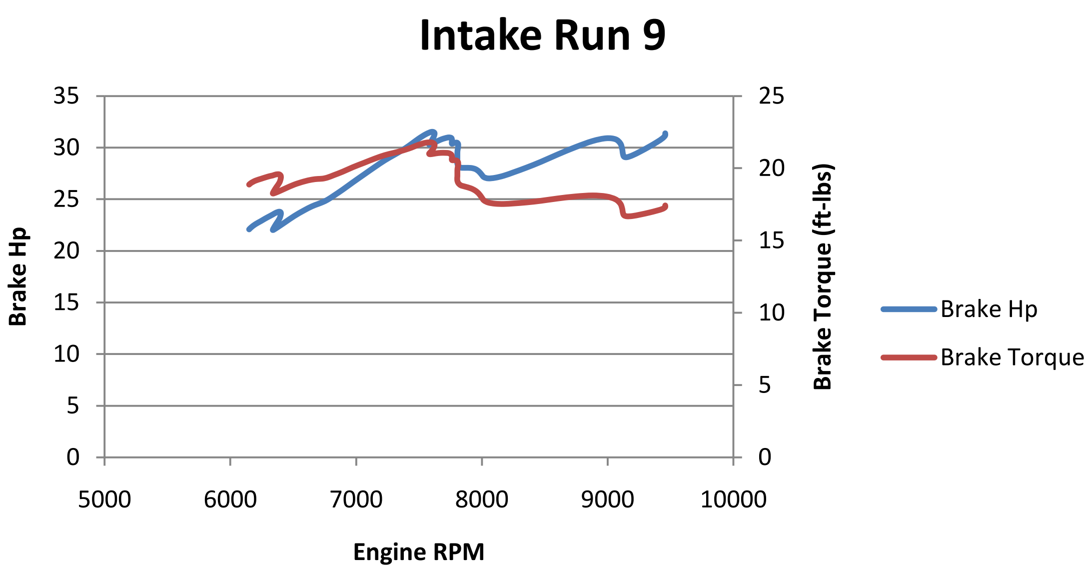
Configuration - Run 7
Port - Short Bellmouth Length = 7, 3/4", short Plenum
Short Plenum - Short Bell Mouth
5th gear - 30 Tooth Sprocket



Configuration - Run 8
Port - Short Bellmouth Length = 6, 1/2", Medium Plenum
Medium Plenum - Short Bell Mouth
5th gear - 30 Tooth Sprocket
70% Throttle



Configuration - Run 9
Port - Long Bellmouth Length = 6, 3/4", Medium Plenum
Medium Plenum - Long Bell Mouth
5th gear - 30 Tooth Sprocket

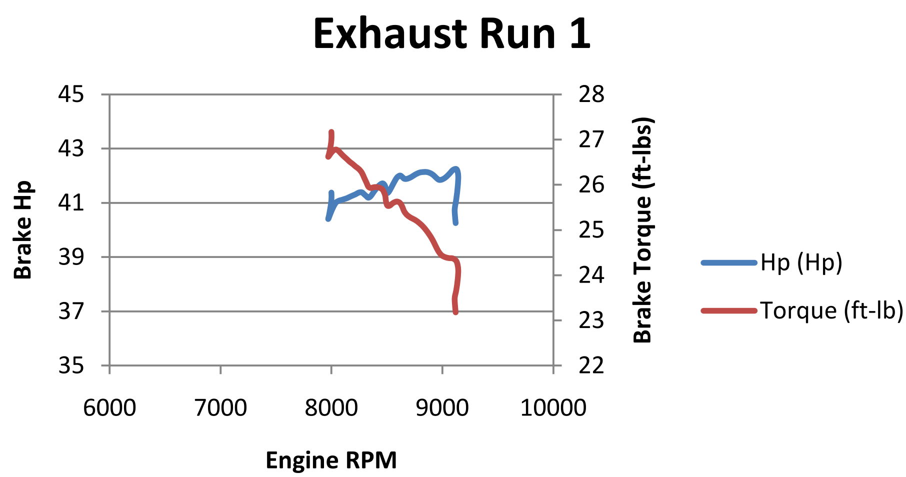


F.2 Individual Exhaust Test Results

Configuration - Run 1
Short Bell Mouth - Port to Mouth
6,1/8"

Short Length

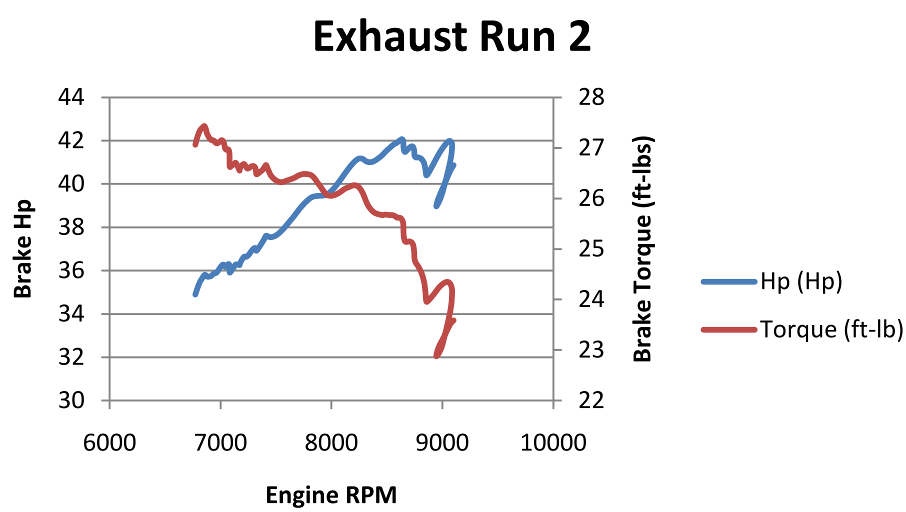
50% Throttle



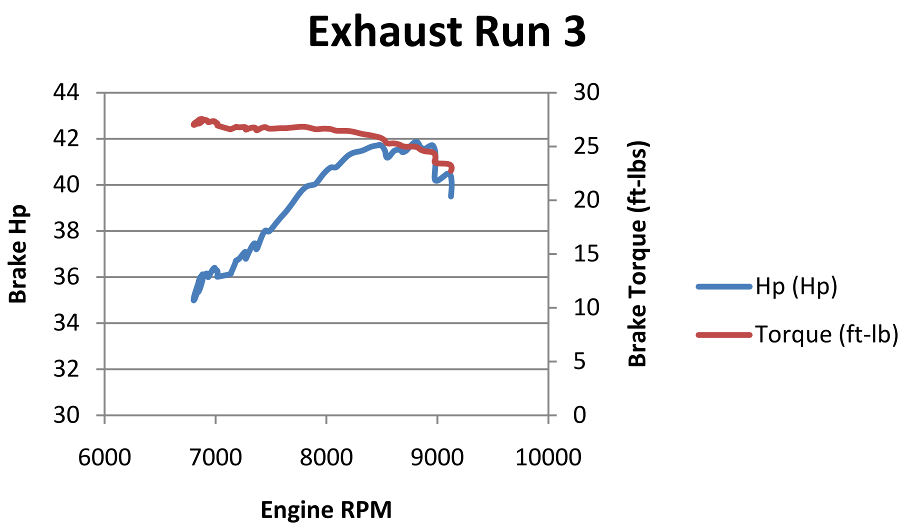
Configuration - Run 2
Short Bell Mouth - Port to Mouth
6,1/8"

Short Length

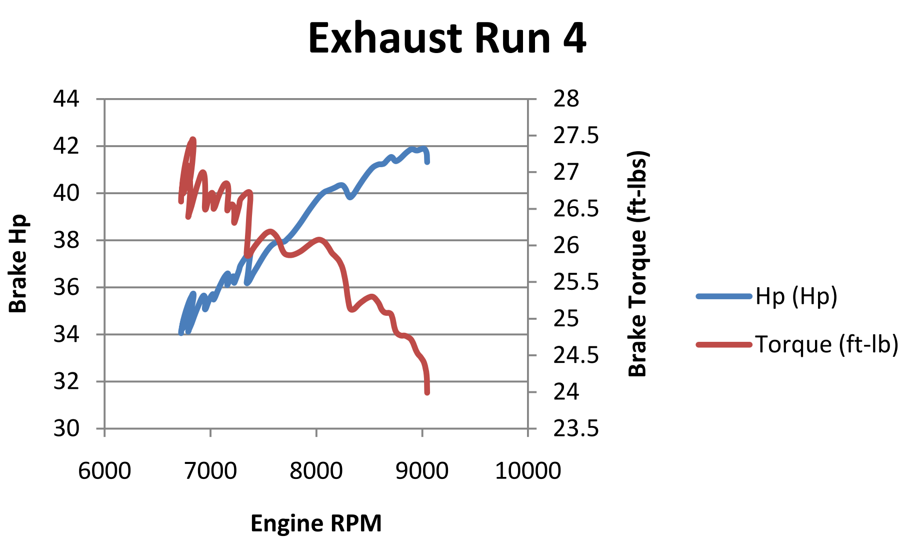
70% Throttle



Configuration - Run 3
Short Bell Mouth - Port to Mouth
6,1/8"
Long Length
50% Throttle



Configuration - Run 4
Short Bell Mouth - Port to Mouth
6,1/8"
Long Length
50% Throttle

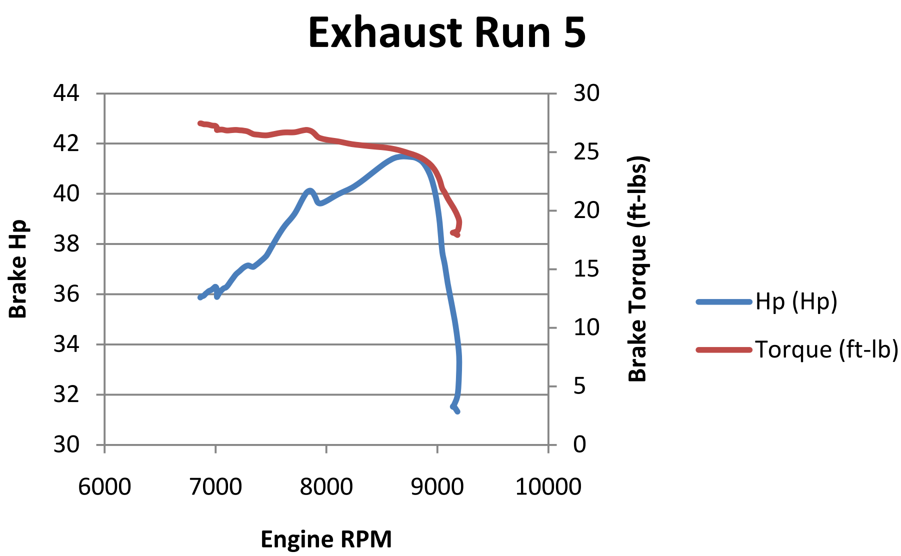


Configuration - Run 5

Short Bell Mouth - Port to Mouth
6,1/8"

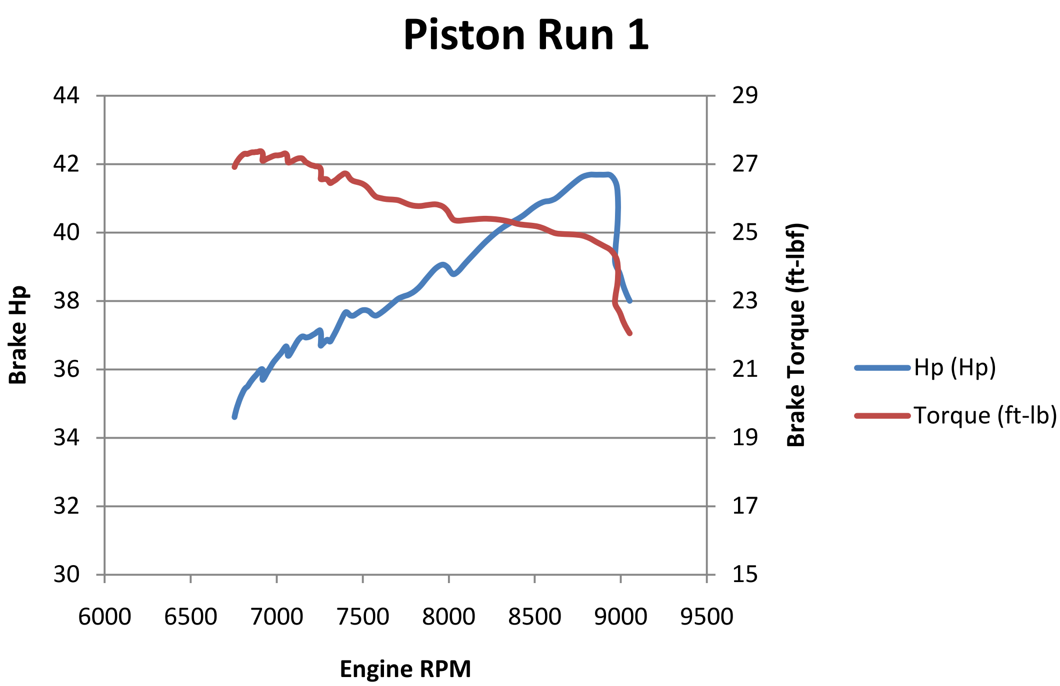
Step

50% Throttle

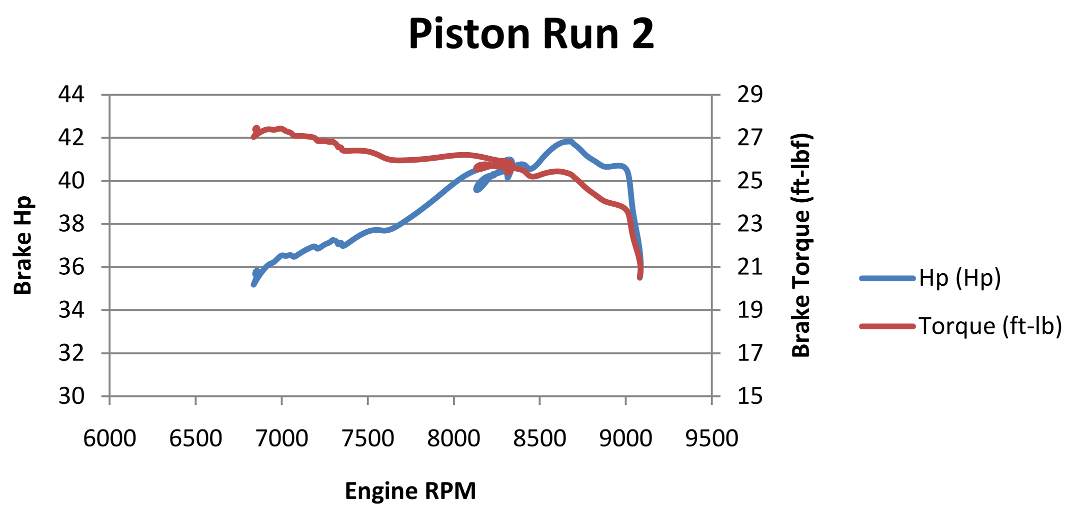


F.3 Individual High Compression Piston Test Results

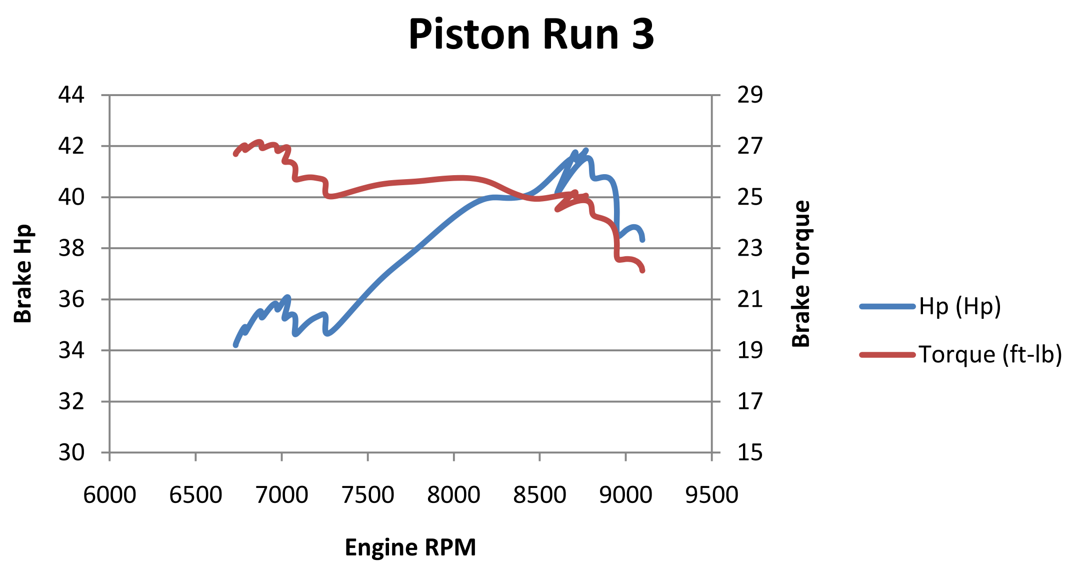
Configuration - Run 1
Medium Plenum, Short Runner - 6, 1/8'
Length
Stepped Exhaust
13.5:1 Compression
50% Throttle



Configuration - Run 2
Medium Plenum, Short Runner - 6, 1/8' Length
Stepped Exhaust
13.5:1 Compression
50% Throttle



Configuration- Run 3
Medium Plenum, Short Runner - 6, 1/8' Length
Stepped Exhaust
13.5:1 Compression
50% Throttle



Appendix G. Exhaust CAD

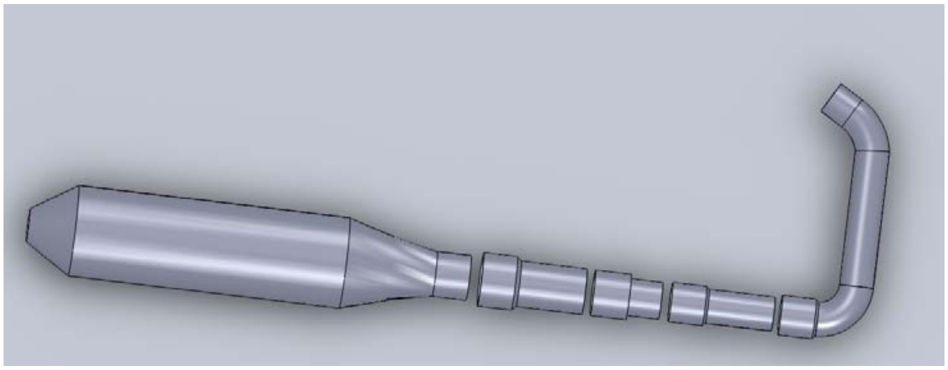


Figure 79: Stepped Exhaust Test Configuration

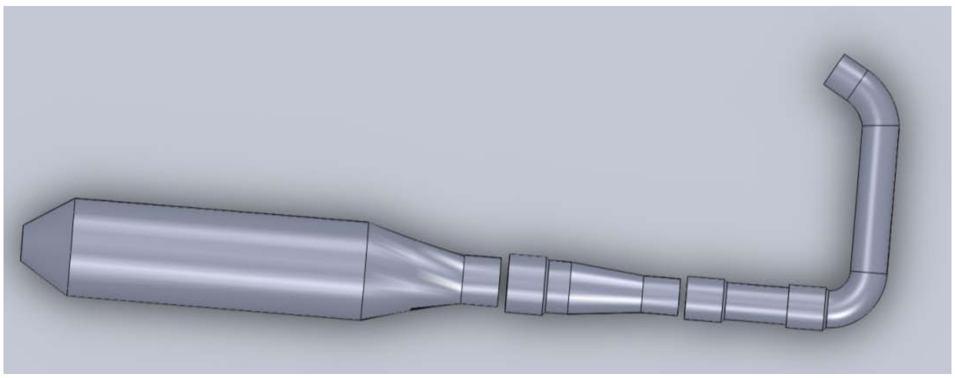


Figure 80: 7,000 rpm Tuned Exhaust

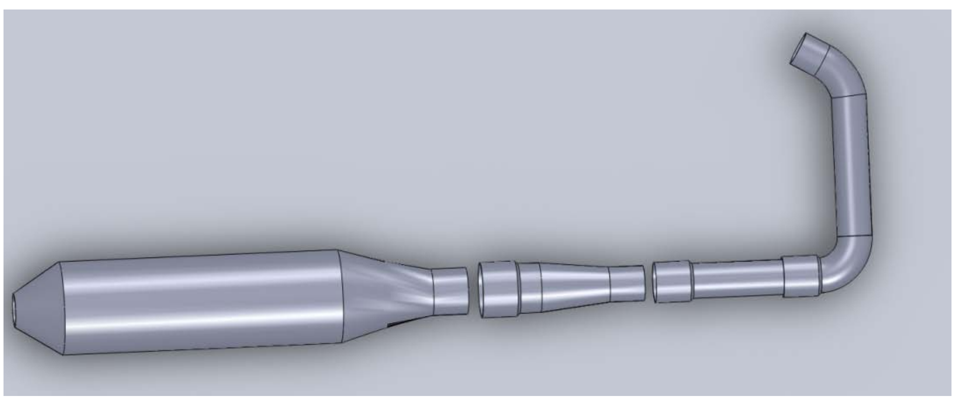


Figure 81: 5,000 rpm Tuned Exhaust

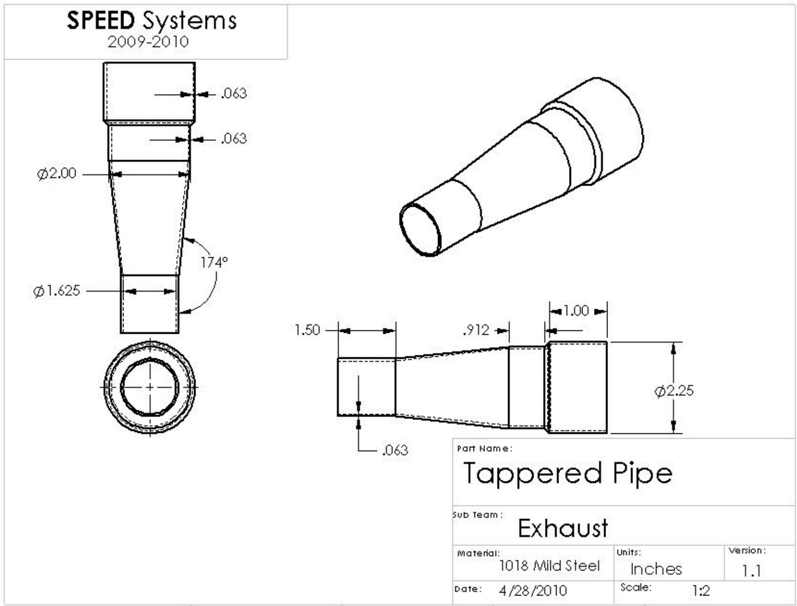


Figure 82: Schematic for Exhaust Megaphone

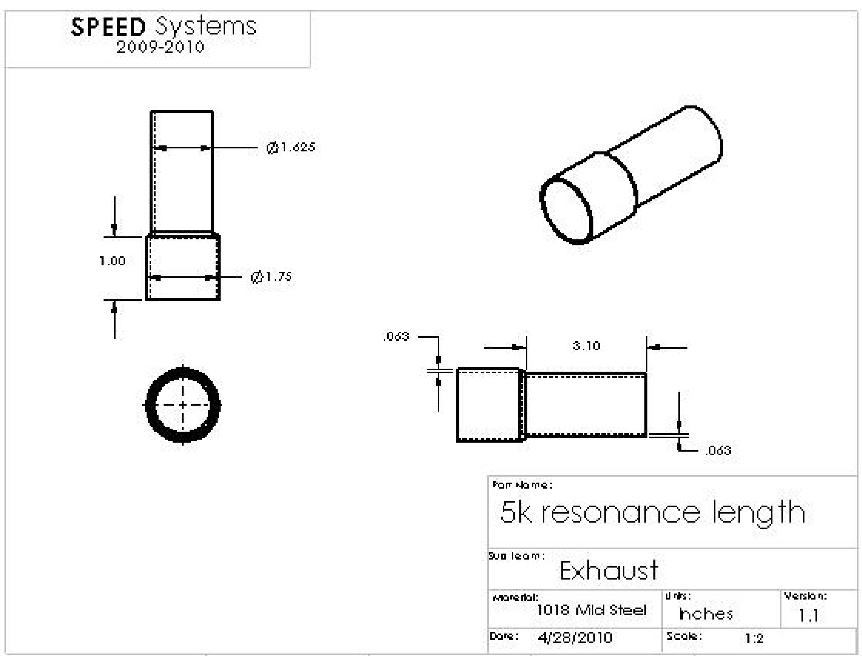


Figure 83: Test Piece for 5k rpm (long) Exhaust Configuration

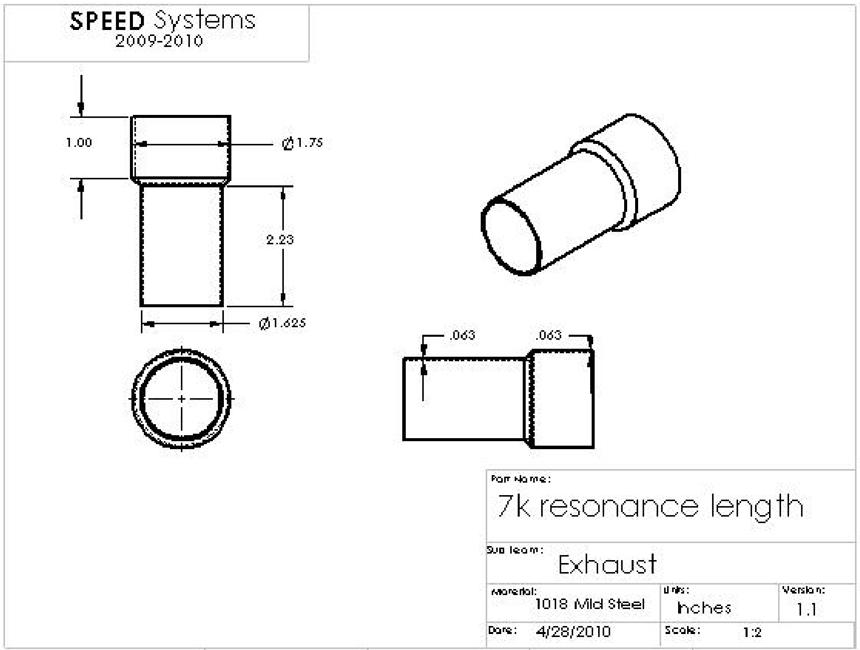


Figure 84: Test Piece for 7k rpm (short) Exhaust Configuration

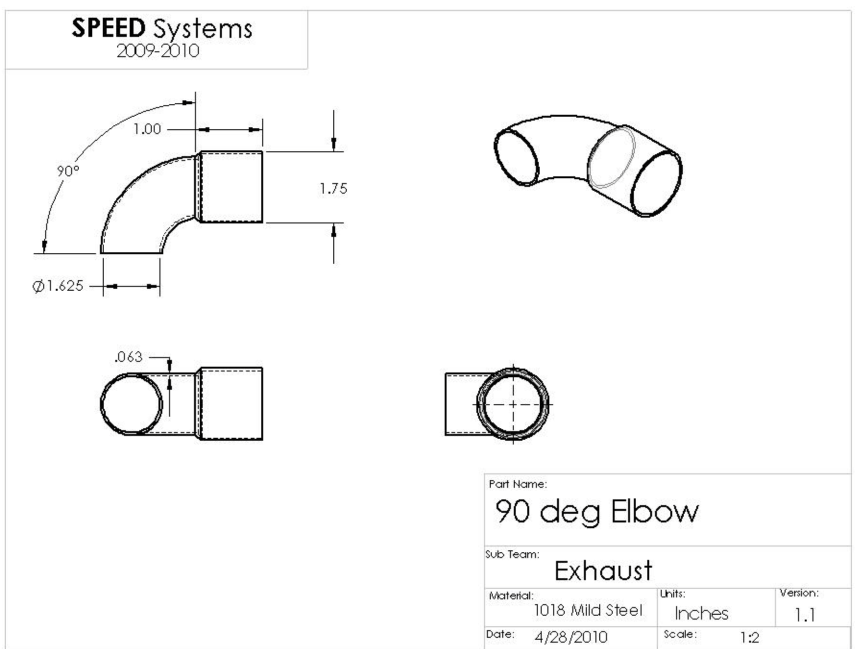


Figure 85: 90-deg Elbow for Exhaust (all configurations)

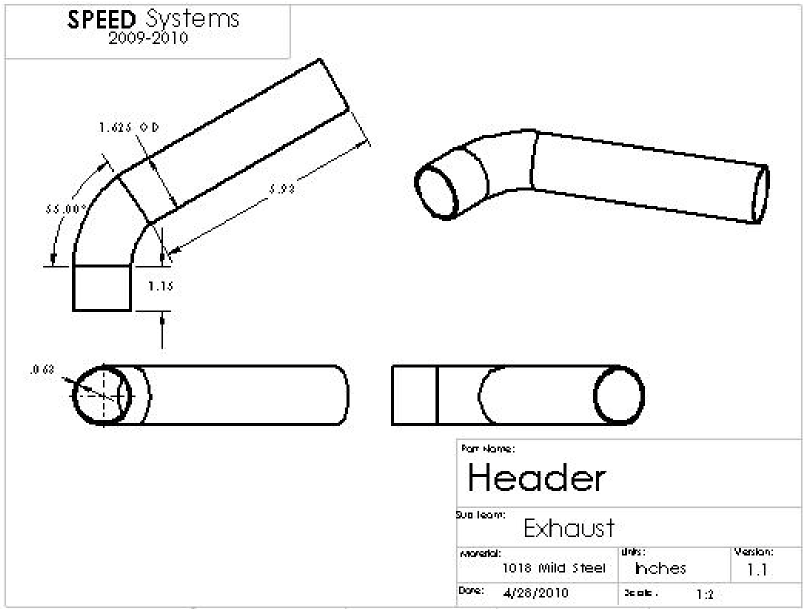


Figure 86: Exhaust Header (all configurations)

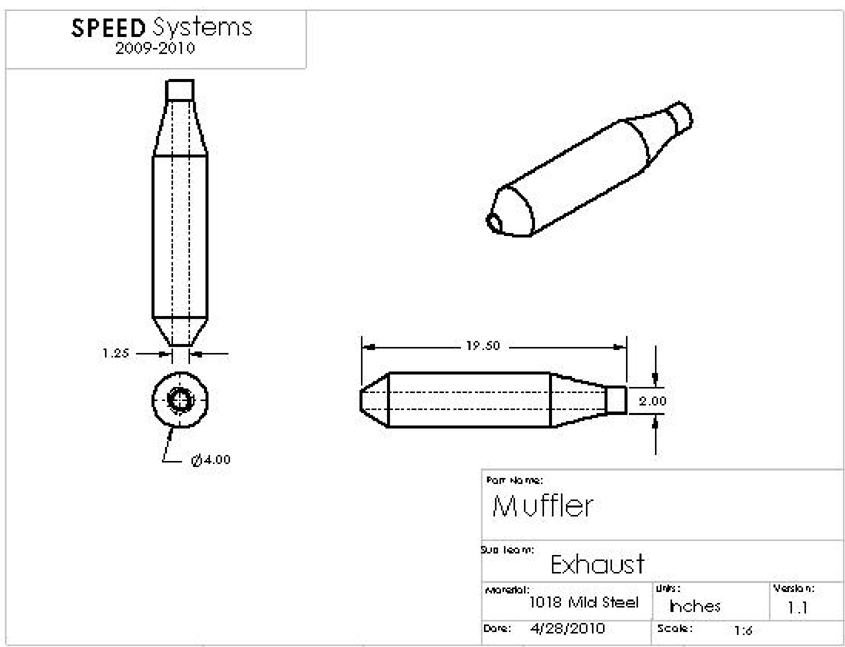


Figure 87: Schematic of FMF Muffler

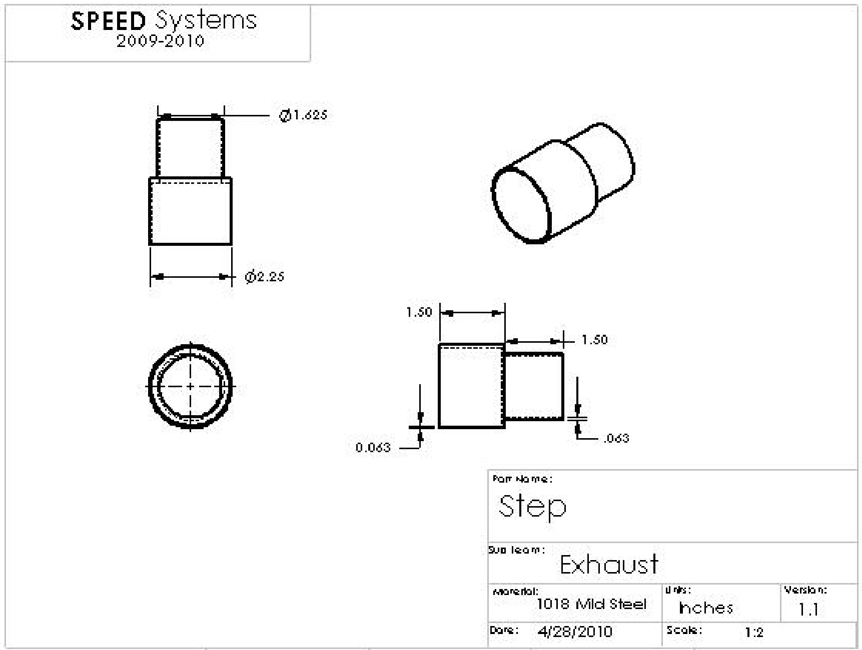


Figure 88: Schematic of Exhaust Step Test Section (step configuration)

Created by Leon Vinokurov

5/31/2010

Ricardo WAVE Tutorial

To access the built in tutorial:

- Open WAVE
- Find *Help* drop down menu above the toolbar
- Click on *Help* and go to WAVE Help…
- Or you can just press **F1** on the keyboard
- Click the WAVEBuild icon
- Select *Tutorials* in the box on the left
- Go to Beginner Tutorial >>> Building an SI model

Leon's Tutorial

This is the method of how the WR450 model was created and what improvements can be made, as well as tips on engine measurements, graph creation and building multi-cylinder models. This tutorial is based on the built in help in WAVE.

You can follow along with the WAVE Tutorial while using this one, to gain a bigger, more in-depth understanding of what is going on.

If questions remain, consult the Help menu in WAVE or contact me.

*Drop down menus in *italics*

Selections are in **bold

Step 1: Measurements

- Before proceeding, make sure you have all the necessary measurements taken from the physical engine
- Required measurements:
    - Find the bore, stroke, connecting rod length and compression ratio of your engine
    - Intake and exhaust lengths, diameters, as well as bend angles
        - Don't forget the cylinder head! You need (approximate) measurements of intake and exhaust tract length, diameter and bends from the entrance of the head to the combustion chamber
        - Restrictor location
        - Valve diameters
    - Include all step changes in diameter (restrictor modeling discussed later)
    - Location of fuel injector in intake tract
    - Chambered mufflers are more complex to model, and WAVE shows how it is done, but if a straight through muffler is used a straight pipe is a good approximation
    - Camshaft profile and timing measurements are also needed
        - This may be the most challenging part to get right, as engine performance highly depends on valve timing
        - You must measure lobe profile, and figure out the valve events in terms of degrees before/after TDC (top dead center)
        - This can be done crudely by measuring max lift and estimating valve opening and duration
        - A more accurate measurement involves using dial calipers and a degree wheel on the crankshaft to measure valve lift vs. crank angle
        - Try to have your measurements in MS Excel format
    - The finer your measurements are, the more representative the model will be of the actual engine

Step 2: Settings

- Open **WAVE**. It should be provided by the ME department in the ME labs on campus.
- Initial settings: go to *Simulation* and select **General Parameters**

- Define your unit system.
    - Select the units that you will mostly be working with, as the units for any specific measurement can be specified as English or SI any time. It will just be annoying to go to every input and change the units if your measurements were taken in SI, but the specified unit system is English.
- Set **Simulation Duration** to 30 seconds. This sets system convergence time. WAVE stops the simulation after it converges, so this just sets a time that is long enough for convergence. Don't worry about the yellow highlighting WAVE uses here.
- Go to Fuel and Air Properties section and press **TAG**
    - This will allow for selection of fuel
    - If the desired fuel isn't listed, your own file can be created by clicking **Create Properties File**
    - You can set the air composition as well as fuel composition there
    - If the desired fuel's properties are known, press **Create** in the Fuel Input Data box
- Close General Parameters window
- Go to *Simulation* and choose **Title**
    - Pick a name for your model and press **OK**
- **Save** the model

Step 3: Time to begin modeling your engine in WAVE

a. Junctions and Ducts

- Tips on working on the canvas: Click the middle mouse button to move elements already on the canvas, roll the middle button to zoom
- Click and drag **Cylinders** from elements list onto the canvas
    - This puts a single cylinder into you model
- Next, you need to drag and drop **Orifice** elements onto the canvas
    - Orifice elements signify a change in tract profile
    - Think of the Orifice element as a connection between two pipes, representing, for example, a change in pipe diameter or bend angle

- Select and drop as many Orifice elements as deemed necessary by your measurements
- Drag and drop two Ambient elements from the elements tree
    - Place the one representing the intake furthest left and exhaust furthest to the right
    - These represent the conditions of the intake and exhaust gasses
- Connections of multiple tracts into one collector (such as the intake tract going into 3 valves) can be accomplished by the use of **Y-Junctions** in the elements tree
    - Expand Y-Junctions and drag **Simple Y-Junction** onto canvas
    - Define the diameter of the junction/collector in the **Simple Y-Junction** tab
    - Go to **Edit Openings** to configure the junction
    - Set the angle from X and then input the correct angle from Y fopr each duct to properly orient them
- Once all Orifices and Ambients have been placed, connect everything with ducts by simply clicking on the pink dots and dragging the duct to the next orifice
    - Label Ducts, Ambients, and Orifices if desired by double-clicking
    - It is intuitive to go left-to-right, with the left Ambient and associated Ducts representing intake, and the right Ambient and Ducts representing exhaust
- Here is the simplified example from the WAVE Tutorial to show how it should look:

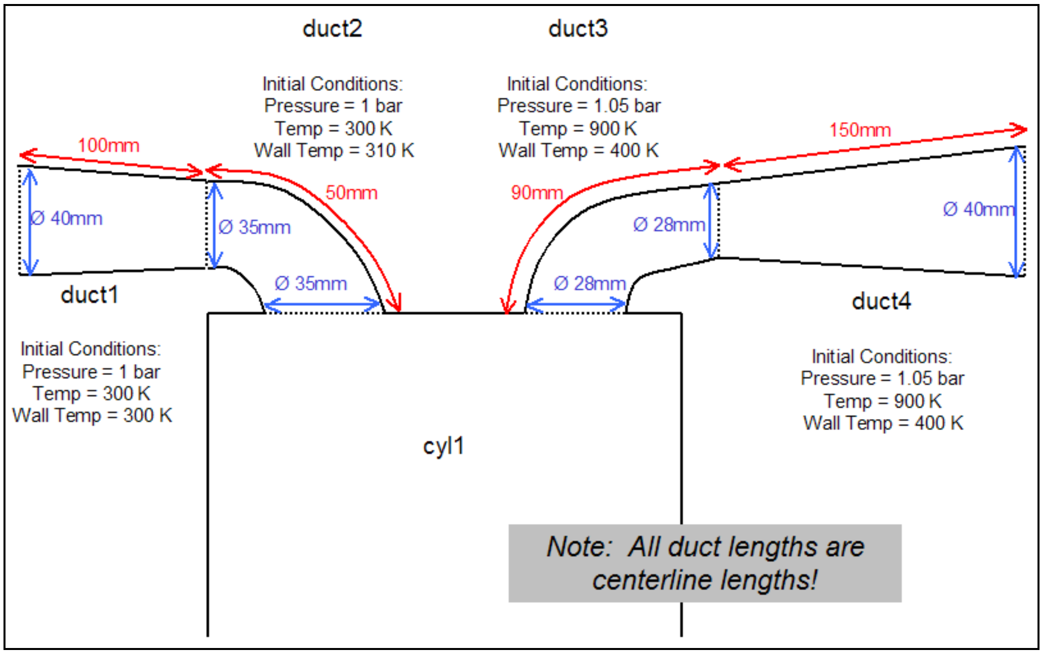


Figure 89: Measurements taken from simplified engine

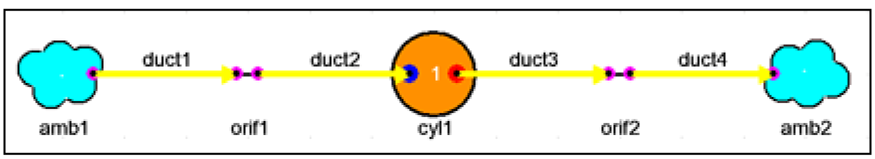


Figure 90: Representation of simplified engine on the WAVE canvas

b. Defining Ambients, Ducts, and Orifices

- Double click the **Ambients** and specify the ambient pressure and temperature
- Double click a **Duct**
  - Under the dimensions tab define **shape**, **left and right diameters**, **total length** and **bend angle**
  - A short discussion on **discretization length** is contained within the Ricardo WAVE Tutorial, Section 3.2
  - Main Point (from Ricardo):

Ricardo's experience in 1-D modeling has indicated a relationship between the discretization length (dx) and the engine bore diameter (B).

For the intake side	$dx = 0.45 * B$
For the exhaust side	$dx = 0.55 * B$

This general discretization technique provides for a relatively uniform model size (in terms of total sub-volume count) regardless of whether the model is a large ship engine or a small, model airplane engine. It is only a general rule of thumb, however. Some refinement of discretization sizes may be required to further optimize the model.

- Go to **Coefficients** tab
    - In Multipliers box, select 0.0 for **Friction** and 1.5 for **Heat Transfer** per WAVE tutorial, Section 3.2
    - Main Point (from Ricardo):

Note that the Friction and Heat Transfer Coefficients have default values of 1.0. These values are multipliers for the standard calculation. Thus values of 0.0 imply that there is no pressure loss due to friction and no heat transfer occurring along the length of the duct while values of 2.0 imply that twice the standard pressure loss due to friction and twice the standard heat transfer is occurring. These multipliers may be used as "tuning knobs" to adjust friction and heat transfer and should be changed according to the surface roughness of the material and flow conditions in the duct. Keep in mind that surface roughness will affect BOTH of these parameters and that pressure loss due to increased heat transfer can be much greater (expansion/contraction of the gas) than pressure loss due to friction!

- Go to **Initial Conditions** tab and change if desired
    - WAVE iterates quickly, so for a naturally aspirated engine, just leave the conditions at their default values
- Once a duct is defined, it should turn black
- The Orifices will reappear once each side is defined

c. Modeling the Restrictor

- There are a few ways to do this, I will mention two:
    - At the location of the restrictor, double-click the Orifice and set diameter as 20mm (or whatever the restrictor size is)

   - Make the ducts leading into and away from the restrictor tapered, with one diameter set as pipe diameter, and the other set as restrictor diameter

d. Define the Engine and Operating Conditions

- Go to the *Model* menu and select **Engine**
   - Input engine parameters into **Configuration** box
   - The Friction Correlation box uses the Chen-Flynn model

$$\mathbf{FMEP = ACF + BCF(P_{MAX}) + CCF(rpm * stroke/2) + QCF(rpm * stroke/2)^2}$$

   - From Ricardo:

The use of the BCF term is to account for changes in P_{max}, which can be used to vary frictional losses across a range of engine loads. The CCF and QCF terms are used to account for changes in rpm, varying frictional losses across a range of engine speeds.

If the simulation is only to simulate tested speed/load points, the FMEP can be entered directly using only the ACF value (directly entering FMEP in the appropriate pressure units) and setting the other coefficients to 0 (zero).

- Go to **Operating Parameters** tab
   - For **Engine Speed**, enter {SPEED}
   - Brackets signify a **Constant** in WAVE
   - Don't worry about the yellow highlighting, it just means the value is currently undefined
   - Click **OK**, and press **NO** when asked to add to constants table
- To Define a constant, either click on the constants icon in the toolbar or go to *Simulation* > Constants > Table
   - Under **Name**, define the constant you have entered
      - In this case, it is called SPEED
   - Then enter all the cases you would like to be tested
      - We're defining engine speed, so enter values from maximum to minimum rpm you want simulated

- Go back to *Model* > **Engine**
    - Select the **Heat Transfer** tab
    - In **Multipliers** box, change the **Piston Top Surface** and **Cylinder Head Surface** multipliers to the measured values
    - Surface area is shown in boxes to the right
    - Further explanation is in the WAVE tutorial, Section 4.3
    - From WAVE example:

Table 18: Example Combustion Model, notice the multipliers

Woschni Model	**Original**
Heat Transfer when Intake Valves are Open	**1.0**
Heat Transfer when Intake Valves are Closed	**1.0**
Piston Top Surface Area Multiplier	**1.0 (flat-top piston)**
Cylinder Head Surface Area Multiplier	**1.6 (to make 7665 mm^2)**
Piston Top Temperature	**520 [K]**
Cylinder Head Temperature	**520 [K]**
Cylinder Liner Temperature	**400 [K]**
Intake Valve Temperature	**420 [K]**
Exhaust Valve Temperature	**480 [K]**
Swirl Ratio	**0.0**

- Go to **Combustion** tab
    - Select **SI Weibe** model form *Combustion Model* drop down menu for a conventional model or choose **Profile** if *Heat Release vs. Crank Angle* data is available

- This models the heat release within the combustion chamber, and essentially stands in for spark timing modeling as it models a rate of burn, duration and a 50% burn point
- A more thorough discussion of the topic is in the WAVE Tutorial
- An excerpt from the Tutorial:

The Profile model is used when Heat Release vs. Crank Angle data is available directly for every speed/load point to be tested by the model. This data can be directly entered into WAVE as a table with a combustion start time and efficiency.

*More widely used, however, the **SI Wiebe** model simply uses an S-curve function that represents the cumulative heat-release in the cylinder. The first derivative of this function is the rate of heat release. The SI Wiebe model is very commonly used and represents experimentally observed combustion heat release quite well for most situations.*

*Select the **SI Wiebe** option from the **Combustion Model** drop-down menu. Enter **8.0** [deg] for the **Location of 50% Burn Point** and **31.0** [deg] for the **Combustion Duration (10-90%)**. Watch the plot actively update as these values are entered. The default value of **2.0** for the **Exponent in Wiebe Function** is appropriate for most cases. Change it and watch the shape of the burn curve change as well. For this example, **2.0** is an appropriate value and should be used. **Fraction of Charge to Burn** should be left at the default value of **1.0** as well.*

e. Defining Cam Timing and Valves

<u>Cam Profile</u>

- Go to *Model* menu and select **Valves**
  - Click **Add**
- The two most often used valve types in WAVE are **Lift** and **Generic Lift**
- Use the **Lift** type if you have measured the Valve Opening vs. Crank Angle
  - Simply define valve diameter in the **Reference Diameter** box and then press **Edit Lift Profile**
  - It helps to have the lift profile vs. crank angle data in Excel so that it can be copied and pasted straight into WAVE

- o You may have to play with profile and cycle anchors, use the plot in the editor to reference valve timing events to their correct positions
- o From WAVE Tutorial, Section 5.1

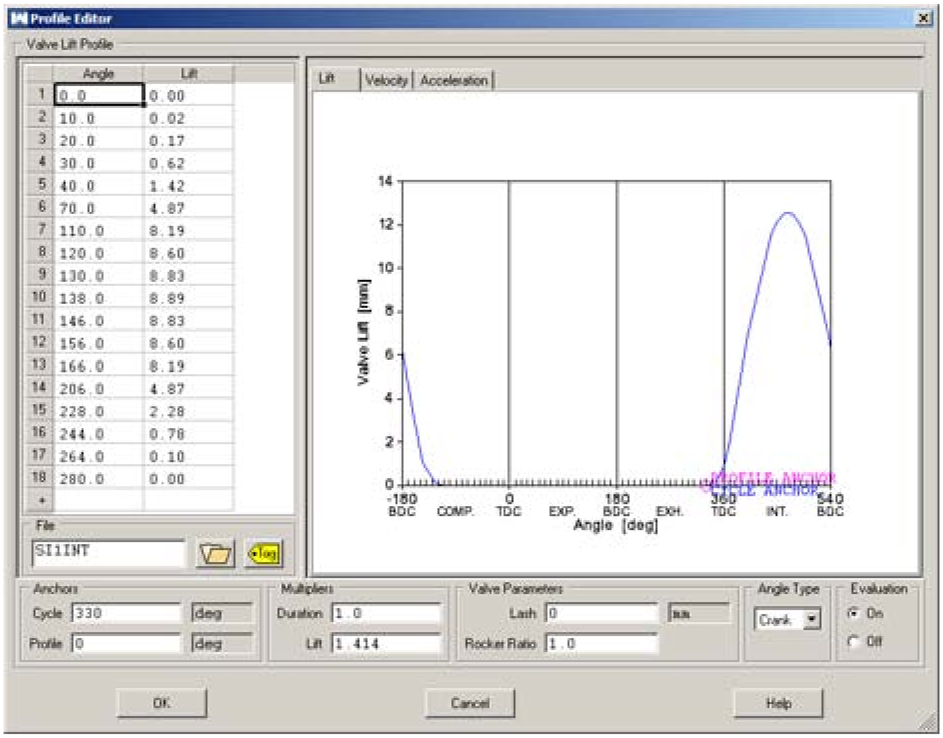


Figure 91: Example valve profile from Ricardo

- Use the **Generic Lift** type if all that is measured is maximum lift and timing events
  - o Input **Reference (Valve) Diameter** and **Maximum Lift** as measured
  - o Select **Edit Lift Profile**
    - ▪ Enter Duration, Lash, and Crank Timing
    - ▪ Crank Timing is referenced to half of total duration
  - o Below, either **S-Curve** or **Polynomial** modeling of cam profile can be entered
    - ▪ S-Curve modeling simply connects the opening and closing ramps of the cam with a specified opening and closing duration
    - ▪ Polynomial modeling is just that, it uses a polynomial curve to model the cam profile
- The **Lift** type is more accurate if the cam profile and timing has been correctly measured

- Otherwise if that data is unavailable, use **Generic Lift** and try to make a good approximation

Valve Flow Coefficients

- Once you are done modeling the cam profile, click **Edit Flow Coefficient Profiles**
  - If flow bench data is available, that can be utilized
  - Otherwise, select **TAG** and choose a flow profile
  - See WAVE Tutorial, Section 5.3 for detailed explanation
- Press **OK**
- Select **Add** to input more valve profiles into WAVE
  - It can store as many profiles as desired

Specifying Valves Within the Model

- Pay attention to the color of the dots that connect the intake and exhaust tracts to the cylinder!
  - BLUE means intake valve, RED means exhaust valve
  - These can be changed by double-clicking on the cylinder icon on the canvas and selecting **Type** within the **Engine Valves** tab
- To specify valves in your model double-click the **cylinder** icon on the canvas
  - In *Engine Valves* tab, click the **Type** menu and select either intake or exhaust, just keep track of which duct goes to which valve
  - Select the profile of the desired valve by using the **Valve #** menu
  - Once all valves in the cylinder are specified, click **OK**

f. Fuel Injectors

- WAVE does fuel injection in two ways:
  - Setting the AFR
  - Setting the mass of fuel injected
- In my WR450 model, I used a **Pulse Width** Injector
  - It is based on the fuel map of the actual engine

- Go to the elements tree and expand the **Injectors** elements
  - Drag and drop a **Pulse Width** injector onto the canvas, near the injector location of the actual engine
  - Connect the injector to the duct which houses it by left-clicking the injector and dragging to the duct
  - Double-click injector
    - In the *Operating Point* tab set **Pulse Width** as {PW} signifying that it is a constant and will be entered into the constants table
    - **Synchronize** to cylinder 1, in drop down menu
    - Go to **Position** tab and place the injector in the measured location
    - In the **Properties** tab, the mixture temperature, nozzle diameter, liquid fraction evaporated after injection and spray spread angle can be adjusted to match reality
    - Go to the **Profiles** tab, and select **Injector Fuel Delivery**
    - This brings up a plot of Mass Injected vs. Pulse Width
    - Calculate the mass injected from knowing injector "size" and pulse width (remember: fuel pressure affects rated injector size!)
    - Input Injection Duration vs. Mass Injected into table
    - Press **OK** to close delivery curve
  - Press **OK** to close the Injector panel

Step 4: Running WAVE and Plotting

- Run **Input Check**  by pressing the icon on the toolbar
  - If anything is wrong, WAVE will let you know
- If it runs through okay, then proceed to running the model  by pressing the **Run Screen Mode** icon on the toolbar
- Once it has finished running, click the Wavepost icon  on the toolbar(it's the next one to the right from the running icons)
  - This will open Wavepost
  - To view output graphs select **Sweep Plots** in the bottom left box
  - This will expand the plots selections, expand **Engine**
  - Choose a subcategory and double-click on the plot you wish to see!

- Here is the finished model of the FSAE WR450 engine in its 2009 configuration

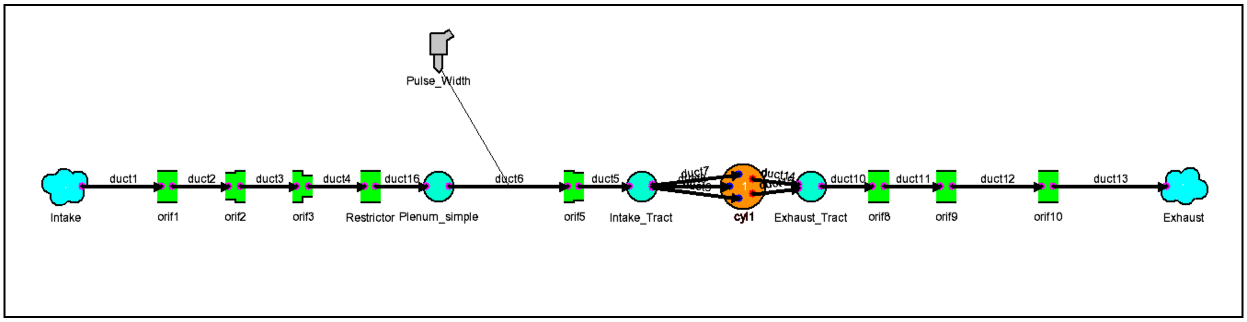


Figure 92: The model for Cal Poly's 2009 FSAE engine, Yamaha WR450

For more information:

My tutorial covers the basics required to get a model up and running in Ricardo WAVE. Use the built in Help in WAVE, it provides a wealth of information.

See the Beginner for extra information, such as making a multi cylinder engine (easiest to copy and paste after you make a working single cylinder model) and creating multi-case sweeps.

The Intermediate Tutorial also goes into some useful information such as **Subcase Sweep Generation**, a very useful feature. If you only do one Intermediate tutorial, **Subcase Sweep Generation** should be the one. You can use it for many purposes. I used it to see which intake and exhaust cam timing would work best.

A quick explanation of 3D plotting in WAVE can be found in the Help menu or my logbook (p. 21).